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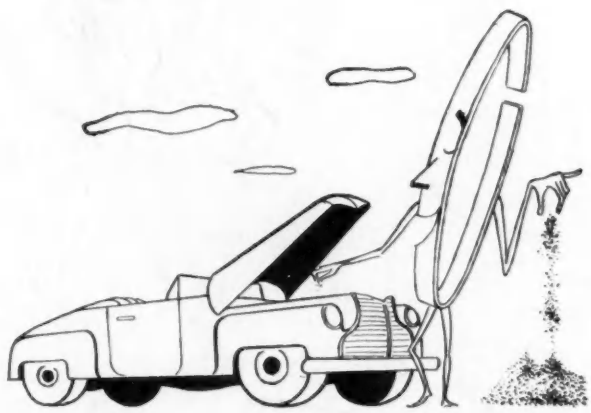

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*Contributed by Stanley J. Szarek, 15 White Ave., Whitesboro, N. Y.**

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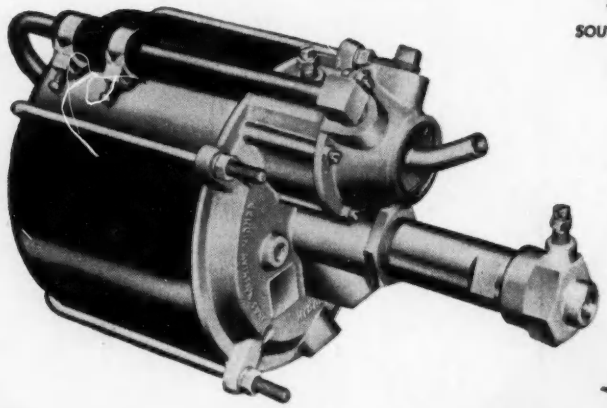

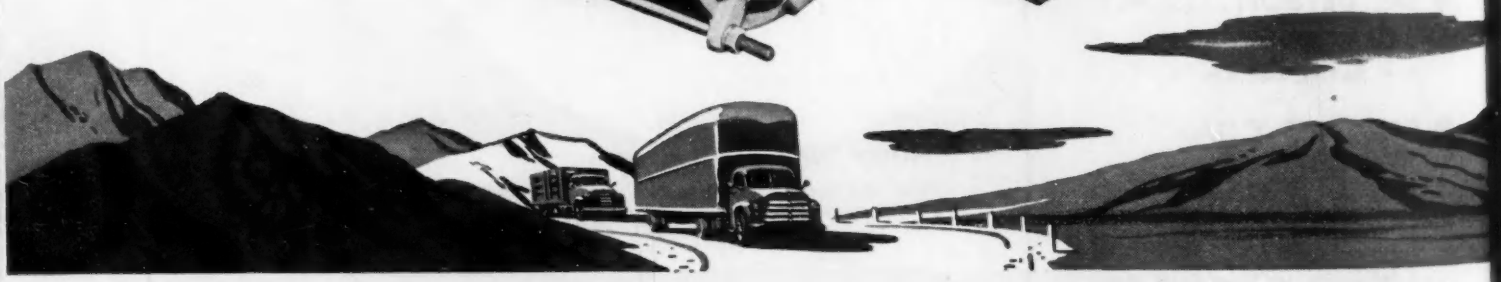

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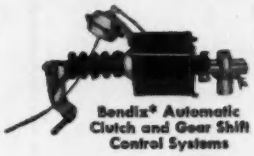

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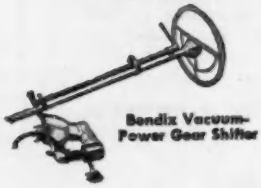

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Air Transport Industry Aim:

ELIMINATION of INEFFICIENCY

EXCERPTS FROM PAPER* BY **R. C. Loomis**
Manager of Inspection & Flight
Consolidated Vultee Aircraft Corp.

SINCE the beginning of 1947, the DC-6, the 202, and the Convair Liner have been placed in regular airline transport operations. The bright outlook for the air transport industry in 1945 and the sincere optimism of the air transport operators and aircraft manufacturers resulted in the ordering of millions of seat-miles of equipment; with the result that in 1948, when most of the postwar transports finally were placed in scheduled operation, the number of available seat-miles had tripled from the year 1944. As the new equipment came into full operation in 1947, the airline picture did not, however, look quite as rosy as it had looked in 1944 and 1945. Instead of substantial profits, all operators were either losing money or were just barely breaking even. It began to appear that the airlines had overanticipated the available market for air transportation because load factors were dropping as the new equipment was added, and as a result of the lower load factors substantial losses were being experienced. Additional new transports were delivered to the airlines in 1948 and the profit picture remained just as bleak as it was in 1947.

In spite of the discouraging financial picture of a majority of the airlines, it would not be forgotten that a substantial improvement in transportation service has been offered to the American and foreign traveling public. When contemplating serious losses, we often forget that, in spite of the tripling of available seat miles, actual passenger load factors still remain at or above the average passenger load factor for the three years prior to the war. Excellent service to the air traveling public probably cannot be offered when the passenger load factors are required to be substantially above prewar load factors, that is, 55-60%. Since the airlines have now placed themselves in the position of offering substantially improved service to the American public, they must reduce their costs so that they can make profits with the equipment they have recently placed in service.

It would appear at first glance that the wonderful new aircraft that Ray Kelly of United Airlines in 1945 looked forward so confidentially to operating were the main reason for the decline of airline profits. Actually, while the costs of introducing new equipment proved to be greater than originally anticipated, there is not a major airline operator in the U. S. that would be in business today if it had not introduced new high-speed, pressurized equipment, because the traveling public demands the service offered by such equipment. Indeed, one major operator, who has now converted to postwar equipment almost exclusively, began to show consistent profits in each of the last quarters of 1948, with every prospect of a black ink year during 1949.

The most significant development in air transport during 1947 and 1948 was the realization by engineers and management alike that inefficient war-

* Paper, "Review of Transport Aircraft Developments during the Past Two Years and Some Predictions for the Future," was presented at the SAE National Aeronautic Meeting, New York City, April 13, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

time design, production, and operating practices could not be tolerated in the economic climate of postwar competition.

Economics

In their drive for a fair share of the air transport market in the postwar era, manufactures and operators have tried to outdo each other by designing and installing new devices to increase safety, to increase speed of operation, and to attract the public's fancy in an effort to obtain maximum sales for that particular operator or manufacturer's product. As a result of the scramble for business, engineers particularly have neglected the necessity for reducing costs through standardization of equipment and accessories. A study of airline stock bins may reveal a vast number of types of altimeters, clocks, pressure gages, gyros, horizons, air-speed indicators, and so forth. The necessity for maintaining parts and supplies for each and every one of these model types can do nothing but increase the cost of the airline operation.

Any discussion of air transport economics cannot disregard the problem of airports of sufficient size and facility to accommodate the number and size of aircraft being introduced in the United States. Many times aircraft designers have proposed aircraft without regard to the economic problem of the community such aircraft must serve. The most economical air transport equipment in the world is of little or no value to the operator unless the communities he desires to serve are financially able to support the size and type of airport required for such aircraft. Tremendous sums of money were spent during the war for the development of military airports, some of which are now available for various communities that operators desire to serve. Communities that inherit supermilitary airports at a very low cost are indeed fortunate, but their neighboring communities simply are not in a position to build such giant airports out of their own funds.

Ground operating costs are a substantial proportion of total airline operating costs. The introduction of aircraft with tricycle landing gears has brought an additional problem in loading rapidly and efficiently during the time the aircraft is on the ground. Also, larger aircraft have a greater fuel capacity, so that methods had to be developed for rapid fueling, as well as for the rapid servicing of mechanical accessories. The variety of aircraft equipment, the large number of airports being operated through, and the number of operators at each airport have imposed a terrific problem of capital investment in ground equipment as well as high operating costs at each airport. In addition, serious problems of equipment congestion at major air terminals have made it vitally necessary to initiate a study for the consolidation of such equipment between operators.

The problem of consolidating airport ground equipment was tackled by the industry on an experimental basis at the airport near Detroit and Cincinnati. An airlines terminal company was formed and, for a time, offered a consolidated ground service to all airlines operating through these terminals. Because airline employees were reluctant to give up their seniority with their respective airlines, however, it was difficult to staff the new company with employees that had the necessary background and experience to make this experiment a complete success. The most successful consolidation of airport organizations is being worked out by agreement between individual airlines.

For example, Braniff and Continental Airlines have worked out consolidated arrangements at seven of the points that they serve jointly. Similarly, Braniff has an arrangement with Chicago and Southern at one station and with Midcontinent at two other stations. Other notable consolidations have taken place between American Airlines and TWA, American Airlines and Continental, and there are probably others.

The control of flight operating costs is even more important than the control of ground costs. In fact, a considerable savings in flight operating costs is possible by careful flight planning, as well as by giving serious consideration to reducing the present tendency toward carrying excess fuel for every conceivable emergency that has occurred to the airlines for the last 15 years. It not only costs a considerable amount of money to operate the aircraft at a high gross weight because of heavy fuel weights, which are never used at the end of a normal trip, but the addition of such fuel almost invariably displaces valuable cargo. Certainly safety should not be jeopardized in any consideration of reserve fuel, but it seems rather silly for an aircraft to be dispatched to New York, for instance, from a point one hour away and have the aircraft carry sufficient fuel to cruise over New York for 2 hr before landing and then land with 1½ hr of fuel remaining in the tank. It would appear to be better practice to dispatch to the congested area and return to the point of take-off rather than to accept a clearance that estimates a 2-hr delay prior to landing.

Air Traffic Control

The tripling of available seat miles of air transportation in the United States between 1944 and 1948 emphasized more than ever the extreme necessity for improving airline scheduled reliability. The greatest single factor affecting airline schedule reliability is the traffic control bottleneck during adverse weather, particularly as aircraft after aircraft arrives over a congested terminal station and is unable to land because of weather being below the acceptable minimum or because landing control systems are unable to handle more than 50% of the arriving aircraft.

In order to provide the ultimate in safe, efficient, air traffic control, reliable automatic devices will, of necessity, have to be designed and installed in the aircraft and used both during contract as well as during instrument weather. The manufacturers of controls are well cognizant of these problems and have already started development of, and in some cases have flying, equipment that will satisfactorily provide the automatic operation required to implement the system outlined by two CAA authors[1].

Prior to the last war, the average traveler came to regard air transportation as a superluxury means

[1] "Effect of RTCA Program in Obtaining Regularity in Air Operations," by D. W. Rentzel and F. B. Lee. Presented at SAE National Aeronautic Meeting, Los Angeles, Oct. 7, 1948.

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of travel. He came to expect from the airlines as a matter of course superlative personal service as well as the extra comfortable seats and a minimum of noise in the cabin. During the war the competition for passenger seats was so great that the air traveler was invariably kicked around by the staff of people catering to his needs. In the postwar era, as the competition for seats decreased, it became more and more evident that it was necessary to re-introduce some of the personal handling aspects so well associated with prewar passenger handling in order to continue to obtain a satisfactory share of the business. Numerous papers have been delivered during the past two years covering the subjects of passenger handling, the design of seats for maximum comfort, and the description of heating and ventilating systems, as well as ways and means of reducing cabin sound level.

The modern supercharged aircraft now in general use in the United States all incorporate supercharging systems that, when operating satisfactorily, give a high degree of passenger comfort, and almost all of the designs allow cooling of the cabin during high atmospheric temperatures. The industry, however, has been plagued with operational difficulties in these supercharging systems. It is hoped that the experience gained in the last two years of operating ths new equipment will allow its effective and continuous operation during 1949 and beyond.

I believe that one of the most significant improvements during the past two years has been the better service offered the air traveler. No longer does the air traveler normally expect to be delayed on each and every flight, no longer does he expect inefficient handling at the passenger station, no longer does he expect to have his baggage lost or smashed en route. I can say from personal observation and experience on many trips during the last few months that the airline passenger service and the degree of passenger comfort offered by the airlines have both become equal to or vastly better than it ever was prior to the war. I think that this is a great tribute to the transport industry, which has been able to provide this service during its difficult years of readjustment.

Airworthiness Requirements

Airworthiness as outlined by the CAA naturally controls the whole scope of aircraft manufacturing and air transport operation. There is no question that the high standard of operating safety established by U. S. air carriers is a combination of excellent engineering design and a high standard of regulations outlined by the CAA. The achievements of the industry would not have been possible without a controlling hand in the form of Civil Aid Regulations.

Unfortunately, the trend of Civil Air Regulations during the past two years has been to become more and more arbitrary in the setting of rules. The arbitrary requirement of a flight engineer's station and the arbitrary requirement of smoke detectors in the interests of safety are two of many examples of regulations that are opposed to the idea of design for a common objective; that is, cockpit efficiency and control or prevention of fire. Such regulations specifically state that an airplane is unsafe beyond a certain weight without a third member added to the crew or that a specific device is necessary to warn or prevent fire in an isolated cargo compartment. If the Civil Air Regulations continue to deviate from the field of objectivity and tend toward the arbitrary, then the excellent progress that has been made in air transport will, to a greater and greater extent, be stifled.

Air Cargo

One of the important developments in air transport during the last two years has been the greatly expanded use of air freight in the United States and overseas. Immediately after the war enumerable small nonscheduled air carriers started operation. Today many of these operators are carrying quantities of cargo never dreamed of in the prewar era. In addition, the scheduled airlines are carrying substantial amounts of cargo, to the extent that today more cargo is being carried than all the combined freight, passengers, mail, and express, prior to the war. More recently, the operation of the Berlin Air Lift by the United States Air Forces has indicated more than ever before the possibilities of air cargo operation through scheduled reliability and intense concentration in the carrying of materials and supplies over an established route.

Flight Deck Design

I don't suppose that anywhere in an airplane can you get such divergent opinion, as many different ideas, or as many strong arguments, as you can when you discuss the cockpit. Here, in this vital area, everybody is an expert and everybody feels free to outline his own ideas. Cockpit mockup boards have come and gone and still aircraft designers are plagued with a multiplicity of requirements if they have more than one customer to satisfy. Within 100 hr of starting operations with a new aircraft, the airline will begin to redesign the cockpit it had agreed was theoretically perfect.

Recently, however, a valiant effort has been made to standardize cockpit arrangements, especially in regard to instrument panel arrangements. Recent agreements between the Army, Navy, and civil authorities have been reached on optimum arrangements. As the necessity for cockpit standardization becomes more recognized, we can fully expect that the numerous designs and requirements in single types of aircraft will diminish.

The Air Force and the Navy have probably approached the problem of cockpit design more objectively than have the civilians.

Powerplants

During the past two years we have seen the introduction and successful operation of automatic feathering propellers, water injection, reversing propellers, low-tension ignition, commercial application of exhaust turbodriven superchargers, and vast strides in the military use of jet engines.

Some Predictions

I believe that certain basic obstacles stand in the way of the development and use of a jet transport before 1965. First of all, every major airline in this country has either just completed the purchase of new, expensive aircraft equipment, or is in the process of re-equipping with such aircraft. The tre-

mendous capital investment required by this program has, for the time being, exhausted the availability of capital funds. The airlines will have to prove that they are capable of making a profit with this new equipment and amortize a per cent of present expenditures before the equity market will support additional funds. The development of jet transports will be a tremendously costly project and, unless government funds are spent for this development, capital will not be available for several years.

Second, the traffic control problem will be greatly complicated by jet transports because of their inability to operate efficiently within existing traffic control problems. Even the CAA has warned that complete traffic control procedures and equipment for high-speed, straight-in approaches to land will not be in operation until 1960 or 1965. These procedures and equipment are essential before jet transports can be operated efficiently.

Third, the problem of stopping heavily loaded, fast landing aircraft on wet or icy runways has only recently been met by the reverse pitch propeller. If the reversible propeller had not become available for current high-speed transports many schedules now being operated would have been canceled because of runway conditions. Some new means, therefore, will have to be developed to stop jet transports safely after contact with wet or icy runways.

The Navy and the Air Force are on the threshold of flying turboprop-equipped aircraft that promise startling performance in conventional-type aircraft. Through the military operation of complicated turboprop controls it is entirely likely that satisfactory and reliable controls will be developed that will make the turboprop practical for commercial transport operations within the next three years.

Turboprop powerplants can be installed in existing transports to allow greater payload (because of lighter powerplants) and improved performance because of higher available emergency powers and ability to use a higher percentage of emergency power for economical high-speed cruising. The turboprop is easily adaptable to present traffic control procedures and the propeller is still available for reversing upon landing.

I believe that the next significant development in commercial air transports will be the application of turboprops to existing airframes. At the same time accessory systems will be greatly simplified by virtue of the ability of the almost constant-speed turbines to supply high-frequency a-c power, simplified cabin pressurization, and simplified cabin and wing heating through the use of turbine-exhaust heat. Such transports will also probably be converted to pneumatic systems in lieu of the present highly inflammable, messy, hydraulic oil. In any event, noninflammable hydraulic fluids will probably be introduced in the near future.

"A Good Team and a Good Game . . ."

by STANWOOD W. SPARROW, President, SAE

FROM as far back as I can remember, I have heard of the trials and tribulations of one-armed paper hangers. Sometimes a member of our own Society is inclined to wonder if the SAE should not have followed the one-armed paper hanger pattern.

If, for example, truck transmissions happen to be the source of the member's daily bread and monthly paycheck, he may feel that all SAE activities should center around gears, and clutches and shafts. He may even begrudge time devoted to passenger car transmissions, considering them but puny toys masquerading under the name transmission. And he may actually grow violent at the thought of precious moments being spent on a subject as fantastic as detonation.

All this may happen, but it seldom does for SAE members have adequate mental machinery and if they put it to work they soon realize that the effectiveness and strength of the SAE resides not in the fact that it includes members interested in a particular type of automotive equipment, but rather that it includes members interested in all types. This West Coast Meeting is a good example of what can be accomplished by a group of activities working together.

This inclination to wonder about SAE matters is altogether praiseworthy provided it is accompanied by a little serious thinking and investigation.

A person with a pain in his big toe isn't in the mood to appreciate the wealth of medical knowledge on diseases of the eyes, the ears, and the throat. Yet if he limps around in an assembly of medical men he is almost certain to find one in the group whose chief interest in life is in the ailments of big toes and when the pain departs he will be grateful for a program of research broad enough to include his own particular ailment. And because of the broad field of SAE activities, any member is almost sure to find that attention is being given to problems that are down his own particular alley and that what is being done is definitely worthwhile.

The SAE was founded by men in the "automobile game" and some of those pioneers were qualified to play any position in that game. The "automobile game" has long since given place to a gigantic automotive industry and while we may still call it a game it isn't a small game or a simple one. What is more, a lifetime is now hardly long enough to provide the skill and knowledge essential to playing one position in that game to the best advantage. Yet the better a player understands the game as a whole, the better he can play his own position and the more fun he will have. The structure of the SAE has been shaped to meet this need. It helps the member meet the problems in his own particular field and it keeps him aware of problems in allied fields and how they are being solved.

Your Society has grown, but its members have grown together not apart. We play different positions but we are on the same team and it is a good team and a good game.

Excerpts from dinner talk at SAE National West Coast Meeting, Aug. 16, 1949. For a full report of the meeting, turn to page 66.

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Which Way to MORE ENGINE POWER

• *Higher Compression Or Supercharging ?*

EXCERPTS FROM PAPER* BY

Max M. Roensch
Research Coordinator
ETHYL CORP.

AUTOMOTIVE engine design can move toward greater performance via three routes—increased compression ratio, higher volumetric efficiency (supercharging), or a combination of the two. Supercharging permits the engine to subsist on a lean octane number diet; upping compression ratio nets better fuel economy. But both demand sturdier engine design.

If engines will progress along the lines of presently known designs, the relationship between antiknock quality and compression ratio may be as shown in Fig. 1. This curve clearly indicates that compression ratios cannot continue their upward trend unless some method is found to produce gasoline of high road octane number at a reasonable cost, or some changes are made in engine design to reduce the octane number requirement at a given compression ratio.

Kettering[1] pointed out the advantages of using high compression engines; and the experimental engines built by GMC Research Laboratories Division demonstrated that high compression engines

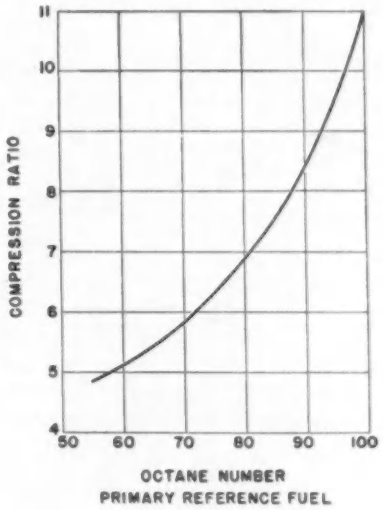

Fig. 1—Effect of compression ratio on octane number requirement

[1] See SAE Quarterly Transactions, Vol. 1, No. 4, October, 1947, pp. 669-679, "More Efficient Utilization of Fuels," by Charles F. Kettering.

* Paper "Engine Design for Past, Present and Future Fuels," was presented at SAE Williamsport Section, Feb. 7, 1949; at SAE Milwaukee Section, March 4, 1949; and at SAE New England Section, Boston, April 5, 1949, and SAE Atlanta Group, Sept. 19, 1949. (This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

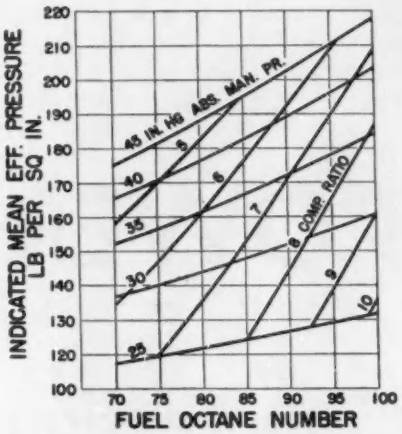

Fig. 2—Effect of compression ratio and manifold pressure on octane requirement and power shown here was determined in laboratory tests on single-cylinder engines

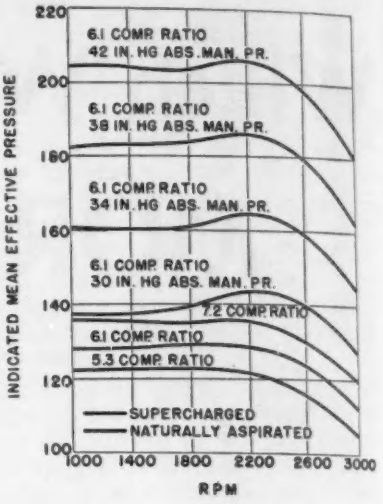

Fig. 3—Notice the difference in engine imep achieved by raising manifold pressure as against that by increasing compression ratio. A separately-driven supercharger with which this overhead valve truck engine was run made possible the boost in manifold pressure

would operate satisfactorily on possible commercial fuels of high antiknock value and give both excellent performance and remarkable gains in fuel economy.

How change in compression ratio affects economy when performance is maintained constant is shown by data from a series of tests conducted by our engineers. The car selected for these tests had a 6.1 compression ratio and a 4.4 rear axle ratio. A series of tests at constant speed were made to establish the level road economy of this vehicle. The acceleration was checked to determine the performance in high gear. Then the compression ratio was increased to 8.5 and the rear axle ratio reduced to 3.9.

Performance checks showed that the acceleration from 10 to to 30 mph was practically the same. Average fuel economy at constant speed over the speed range from 10 to 70 mph was increased from 17.6 to 21.2 mpg, an improvement of 20.6%.

Then the car was put on the road with these two combinations to see what improvement in economy might be expected by the average owner. The test course combined city, suburban, and country driving. The high compression combination increased the tank mileage from 14.3 to 16.7 mpg, a 17% improvement.

Forecast 30% Mileage Gain

Increasing compression ratio does not improve fuel economy while brakes are being applied or while idling at a stop light. Because of these and other conditions of typical car owner operation, the "ideal" figure of 20.6% was reduced to 17%. It is expected that new engines, designed for operation

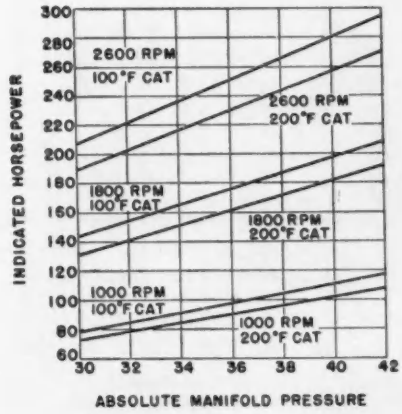

Fig. 4—There is a straight-line relationship between horsepower and manifold pressure, tests with the supercharged truck engine showed. These curves also illustrate the effect of increasing carburetor air temperature

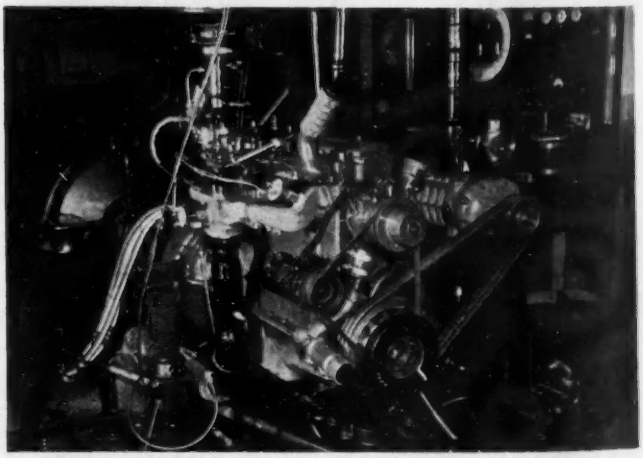

Fig. 5—Laboratory set-up for testing overhead valve truck engine coupled to a Roots blower

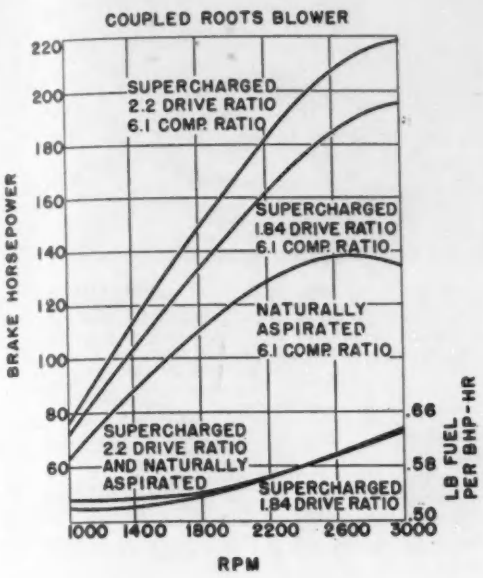
COUPLED ROOTS BLOWER
BRAKE HORSEPOWER
LB FUEL PER BHP-HR
RPM
SUPERCHARGED 2.2 DRIVE RATIO 6.1 COMP. RATIO
SUPERCHARGED 1.84 DRIVE RATIO 6.1 COMP. RATIO
NATURALLY ASPIRATED 6.1 COMP. RATIO
SUPERCHARGED 2.2 DRIVE RATIO AND NATURALLY ASPIRATED
SUPERCHARGED 1.84 DRIVE RATIO

Fig. 6 (left)—Performance of the supercharged truck engine coupled to a Roots blower

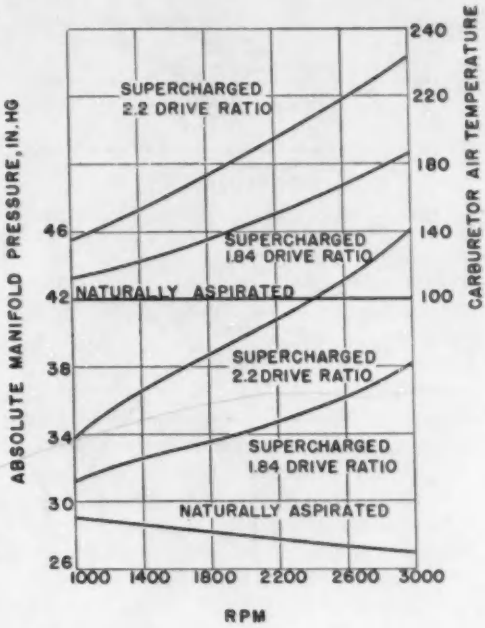

Fig. 7 (right)—Effect of supercharger on manifold pressure and carburetor air temperature

at a compression ratio of 8.5 to 1 and upward and having high mechanical efficiency, will give 30% more tank mileage than present production engines with no sacrifice in performance.

Assuming we are to have higher antiknock gasoline in normal distribution, the problem resolves itself into determining how to use these octane numbers most effectively. Much test work in the Ethyl Corp. Research Laboratories has been carried out on single-cylinder test engines to establish some of these basic principles.

Fig. 2 shows the relationship between compression ratio, manifold pressure, and octane number on a single-cylinder laboratory engine. These data were obtained at 2500 rpm engine speed and the octane numbers are the Motor Method octane numbers of matching blends of secondary reference fuels.

Truck & Bus Engines Tested

Since this engine had a displacement of only 24 cu in., it was felt that similar information should be developed on production automotive engines. Several engines have been tested at various compression ratios, both with and without supercharger. And we recently completed an extensive series of tests on two truck engines—one an L-head and the other an overhead valve design. The following discussion covers the test data on the overhead valve engine, as it is used in both truck and bus service.

The engine was tested at three compression ratios, naturally aspirated, and then with a separately driven air compressor capable of supplying the engine with air at pressures considerably above atmospheric conditions. With the compressor and other allied equipment, it was possible to operate this engine at any manifold pressure up to 42 in. Hg absolute and at any carburetor air temperature from room temperature to 250 F.

Fig. 3 shows the results of some of the tests on the engine with this equipment. Increase in imep with each change in compression ratio is noticeable; but when the engine was supercharged, the gain in output was many times that from increasing compression ratio. These data were obtained with an air-fuel ratio of 12.8 to 1 and a carburetor air temperature of 100 F. This gives some idea of the response of the engine to the changing conditions of increased compression ratio and manifold pressure when all other conditions are maintained constant. It also indicates that when a considerable increase in power is desired from an engine, supercharging is a possible solution.

Fig. 4 shows that within a reasonable range of manifold pressure, the indicated power increased as a straight line function of the manifold pressure at any one speed. Since supercharging tends to heat the air, effect of increasing air temperature also is shown in Fig. 4. These data permit an estimate of the increase in power that might be expected with supercharging.

To check our own calculations and at the same time get actual performance data on this engine, a series of runs was made with commercial supercharger of the Roots type. Fig. 5 shows the engine with the coupled supercharger as it was used in these tests. This was not designed to be installed in a truck, but primarily to make a flexible rig to test different superchargers with several different engine-to-supercharger drive ratios.

Fig. 6 shows performance of the engine with two different supercharger drive ratios. The increase

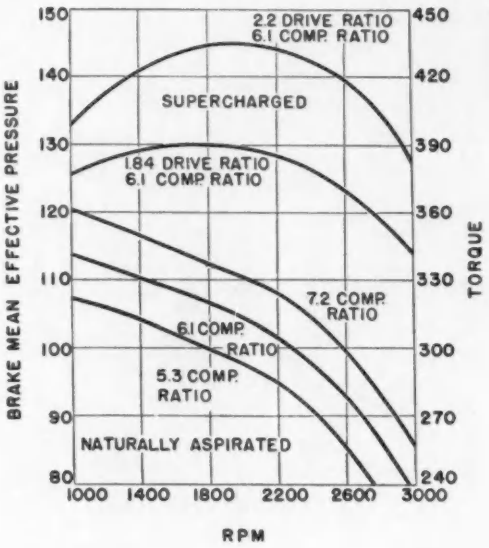

Fig. 8—How compression ratio and supercharging affected both bmep and torque of the overhead valve truck engine

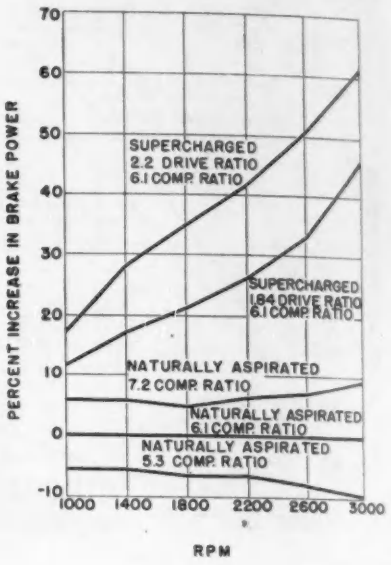

Fig. 9—Raising compression ratio of the truck engine increased power by 9%. But supercharging the engine with a Roots blower yielded a 62% gain in power

in horsepower with a supercharger is immediately apparent. Brake specific fuel consumption data also are shown. An important point to notice here is that there is no appreciable change in specific fuel consumption with this high horsepower.

Effect of the supercharger on manifold pressure and carburetor air temperature is shown in Fig. 7. These data point up the importance of higher supercharger efficiency. Reason: as manifold pressures are increased, the air temperature to the carburetor also increases very rapidly; this higher air temperatures produces a loss in power and an increase in octane number requirement.

Fig. 8 shows the bmep and torque for this same engine and indicates the effect of both compression ratio and supercharging. The percent increase in horsepower for these various conditions is shown in Fig. 9, based on the standard compression ratio of 6.1 to 1, naturally-aspirated or unsupercharged. Top gain in power by increasing the compression ratio to 7.2 to 1 was 9%; but the gain with the supercharger was 62%.

These data all were obtained with a gasoline composed of 50% catalytically cracked base stock, 50% commercial iso-octane, plus 3.0 cc tetraethyl lead; it had an octane rating of 100 (Motor Method).

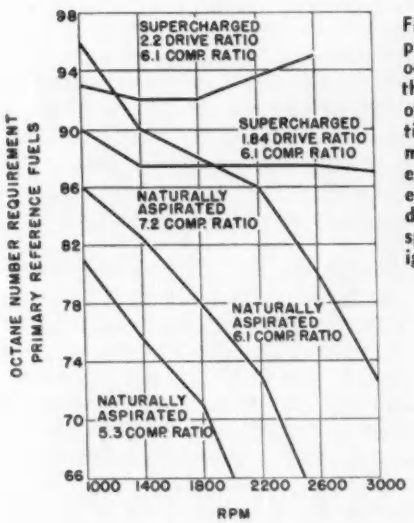

Fig. 10—Tests of effects of compression ratio and supercharging on octane number requirement showed that the octane number requirement of the supercharged engine is practically constant. Octane requirement of the high-compression head exceeded that of the supercharged engine at the lower speeds and then dropped off at the upper end of the speed range. During these tests, ignition timing of the engine was set for maximum power

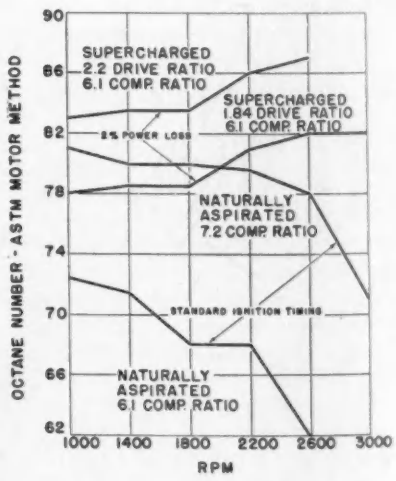

Fig. 11—What the test truck engine would require, both naturally-aspirated and supercharged, in typical commercial gasolines with a sensitivity of eight octane numbers

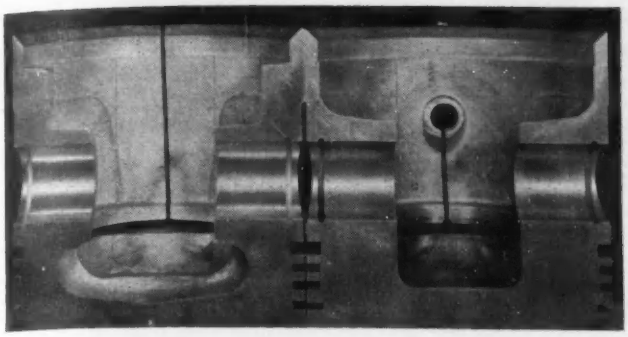

Fig. 12—Modifications for a high output engine incorporated in the piston at right, as contrasted with the one at left, included improved piston head and skirt strength, and redesign to permit use of a 3/32-in. chrome-plated top compression ring instead of a regular 1/8-in. cast-iron ring

The $64 question for the engine designer and even more so for the vehicle operator is: How much of this power is available with present or future gasolines?

To get this answer, the octane number requirement of this engine was determined for the above conditions and these data are shown in Fig. 10. These are the fundamental octane number requirements of the engine in terms of primary reference fuels, which are mixtures of iso-octane and heptane. All data were obtained with an air-fuel ratio of 12.8 to 1, 100 F inlet air temperature, and with maximum power spark advance.

Octane Units Wasted

Effect of increasing compression ratio on the octane number requirement was very marked, particularly at low speeds. And it is interesting to note that the maxium octane requirement of the high-compression head is above the supercharged engine requirement. Also important is the effect of the high volumetric efficiency on octane number requirement. With the supercharged engine, the octane number requirement is practically constant throughout the speed range. Therefore, there is no waste of octane numbers at high engine speed, such as with naturally-aspirated engines. This means the full octane value of the fuel is used throughout the speed range.

It is well-known that the maximum octane number requirement of an engine is not generally used; but the spark is retarded, particularly at low speeds, to a value which is considered a satisfactory compromise, depending on the compression ratio and octane value of the fuel.

An attempt has been made to establish data on a typical fuel for these various conditions of compression ratio and supercharge of this engine. Fig. 11 shows what this engine would require in typical regular fuels, using a regular production distributor with a reasonable amount of spark retard, which results in only a small loss in performance in the normal operating range.

These data show this engine could be operated satisfactorily at either the 7.2 compression ratio, or with moderate supercharging, using present premium fuels. The loss in performance under the two supercharged conditions is 2% throughout the speed range; but with the naturally-aspirated conditions the loss varies with engine speed as follows:

Engine rpm	Percent Power Loss 6.1: 1 C. R.	7.2: 1 C. R.
1000	7	6
1400	2	4
1800	1.5	3
2200	1.5	2
2600	1.5	2
3000	1.5	1

These tests on this full-scale truck engine confirm the tests on the small single-cylinder laboratory engine. They indicate that supercharging yields the greatest improvement in output for a given increase in fuel antiknock value. But there is no reduction in brake specific fuel with this change, since reduced specific fuel consumption comes from an increase in compression ratio, a decrease in engine friction, or an improvement in combustion efficiency.

Thus, for passenger cars the trend will be toward higher compression for more miles per gallon. For trucks and buses it will be either higher compression or supercharging, depending on whether economy or performance is the main objective. Some gain in economy may result from using a supercharged engine with a reduction in axle ratio to maintain the same level of performance.

Withstanding Higher Outputs

Experience has shown that when engine output is increased—either by changes in compression ratio or improvements in volumetric efficiency—modifications are usually necessary in the engine to maintain durability at a satisfactory level. Our experience on some of the engines used for investigating the effect of supercharging gives some indications of changes supercharged engines may require.

Difficulties were encountered with cylinder blocks, pistons, piston rings, and valves . . . parts of the engine most highly stressed, both mechanically and thermally, when engine output was increased 50% or more by supercharging.

In one case the engine manufacturer supplied modified pistons, as shown in Fig. 12, to improve piston head and skirt strength and to allow the use of a 3/32-in. chrome-plated top compression ring in place of a regular cast-iron ring ⅛-in wide. Chrome-plated rings have been effective in reducing scuffing under severe conditions; but the ring problem on one engine was not completely solved until the cylinder block was modified to reduce bore distortion and hot spots.

Engine failures due to preignition during these tests also are interesting. Fig. 13 shows the remains of a piston that failed due to preignition after only 3 min of operation at 2600 rpm, with a manifold pressure of 38 in. Hg absolute.

Cause of this failure was traced to the exhaust valve. Reason for the preignition was the scaling of stem of the exhaust valve head at high temperatures. The valve was of solid stem design with stellite face and was equipped with Rotocaps. Fig. 14 shows this valve after it had been electrolytically

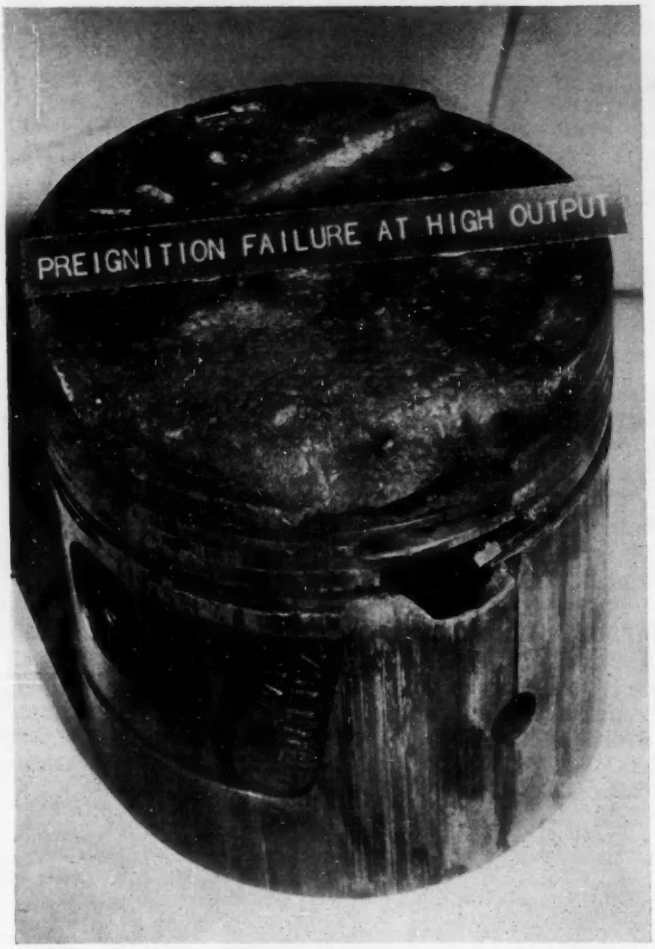

Fig. 13—This piston failed due to preignition after the engine operated only 3 min at 2600 rpm, with 38 in. Hg absolute manifold pressure

cleaned, revealing the chemical attack on the valve head. The iron scale formed was an excellent exciter of preignition.

This failure indicated that a change to either a Bright-Ray exhaust valve head or to sodium-cooled valves are essential for satisfactory operation at 180 psi imep or better.

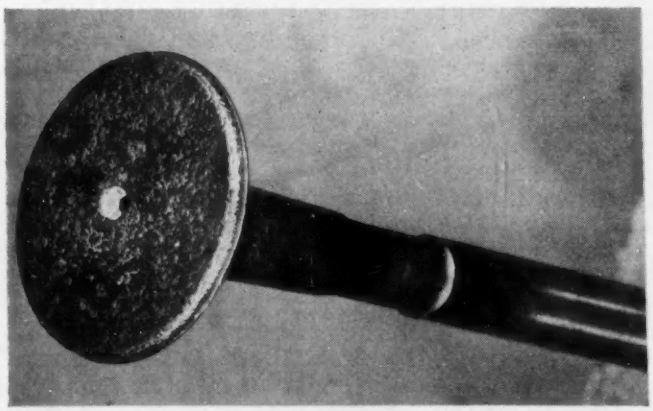

Fig. 14—Scale formed at high temperatures on the head of this steel exhaust valve caused the preignition that led to the failure shown in Fig. 13

The engine was then equipped with stellite-faced, sodium-cooled valves with Rotocaps. Additional lubricating oil was supplied to the valve stem by means of a squirt hole drilled in the rocker arm at the proper angle. This extra oil was essential for proper stem lubrication because of the higher operating temperatures of the valve stem with sodium-cooled valves.

In general, modifications were made on the following parts:

1. Cylinder block—modified cylinder barrel material and cooling.

2. Pistons—reinforced head and skirt.

3. Piston rings—3/32-in. width chrome-plated top compression ring.

4. Exhaust valves—stellite-faced, sodium-cooled.

These engines have operated without difficulty for as long as 420 hr with manifold pressures up to 46 in. Hg absolute—an endurance life that equals or surpasses that of standard production engines operating naturally-aspirated under similar test conditions.

In conclusion, I will outline some changes in engines necessary to give satisfactory durability at high output, particularly in heavy-duty operations as in truck or bus service. Remember that as engine output is increased, either through increased compression ratio or supercharging, peak explosion pressures will be increased from their present level of 500 to 600 psi to 1000 to 1400 psi.

With these increased pressures and the higher temperatures which accompany them, it is felt that future engines will need to have many of the following features if maintenance costs are to be kept down:

1. Rigid well-counterweighted crankshaft with hardened crank pins and journals.

2. Rugged crankcase with cylinders free from mechanical and thermal distortion.

3. Full-length water jackets with water completely around the bores.

4. Water circulation around intake and exhaust valve seats.

5. Valve seats free from mechanical and thermal distortion.

6. Minimum exposure of exhaust valve stem to heat and adequate cooling of valve stem guide.

7. Valve material that has high strength and hardness at elevated temperatures in addition to high resistance to the decomposition products of lead compounds.

8. Sodium-cooled valves.

9. Chrome-plated top compression rings for resistance to scuffing and abrasive wear.

10. Suitable heavy-duty bearings to match the life of the rest of the engine.

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DRAFTING STANDARDS

The SAE Drafting Standards program received prominence at a recent meeting of the American Society for Engineering Education in the form of a paper read by SAE Past-President J. H. Hunt.

The paper points up the philosophy behind the activities of this important SAE technical committee activity and its probable future in relation to national and international projects.

Started early in 1947, with D. G. Roos as Technical Board sponsor, the Drafting Standards project is slated for completion during 1949. Following completion of Roos' term on the Technical Board, A. G. Herreshoff was named sponsor. Chairman of the Drafting Standards Committee is W. A. Siler.

EXCERPTS FROM PAPER* BY

J. H. HUNT

General Motors Corp.

Reason for Project

THE Society of Automotive Engineers some two years ago undertook the development of automotive drafting standards—for two reasons:

First, uniformity of practice within the industry could save some uncertainty and delay by eliminating possible misunderstandings between purchasers and vendors, and simplify the process of transmitting the detailed information from the engineers in the automobile plant to the workman in the parts factory.

Second, various demands exist for unified practice throughout the American manufacturing industry. Military operations today require the use of very technical equipment. A great deal of this equipment is required in such quantities that no single manufacturer could supply all requirements; also, security reasons demand multiple sources of supply. Drawings must be understood by the workmen and inspectors in any plant. In many cases, a factory might have to make something for military purposes which would be quite different from its normal products. Repair work in the field must be carried on by hastily trained mechanics. Uniformity is obviously desirable.

SAE View of Problem

Just as soon as a universally used drafting practice becomes a matter for serious discussion, the question arises as to how to do the job. The practices followed by various manufacturers must be assumed to be meeting the present requirements of each well. It is equally certain that any extensive changes can involve considerable cost, both direct and indirect,

* Paper "The SAE Drafting Standards Program" was presented at a meeting of the American Society for Engineering Education, Troy, N. Y. on June 20, 1949.

and that there will usually be no immediate gains in operations to overbalance the costs in the case of most manufacturers. The universal system should be so planned that the overall cost and annoyance of changes to use it will be a minimum.

This most economical universal drafting practice can be found only after the requirements of each industry are determined. Only after each industry has a clear picture of its own requirements can the system best adapted for all industries be developed. The present SAE program is an effort to establish the facts for the automotive industries. These industries include not only the manufacturers of the passenger cars, trucks and buses usually seen on the highways, but also the manufacturers producing industrial engines and the self-driven equipment using these engines, the makers of completed components, and the makers of the forgings and castings from which complete parts are fabricated.

The difficulty of arriving at agreement and the rather low pressure for such agreement under normal conditions are both illustrated by the experience of one large manufacturer which operates many divisions, under a system of decentralized control. This manufacturer had previously found no compelling reasons for establishing complete uniformity between all of its divisions. It started over two years ago to develop a uniform system for its own use. In spite of the advantage of the aid of a central staff, it has not yet completed its own internal standards, although completion is expected before the SAE project is finished. This experience indicates that a uniform system for all manufacturing industry will be found to be anything but a simple project.

SAE Organization for Work

Recognizing the importance of being prepared to meet all of its own responsibilities in contributing to an ultimate standard, the SAE Drafting Standards Committee has carried on its work with great care. One hundred and eighteen persons organized in 15 subcommittees are working under the supervision of a steering committee of 17, which includes the chairman of each of the subcommittees. With very few exceptions, the men on the steering committee are either chief draftsmen, or men to whom chief draftsmen report. They are thoroughly familiar with today's automobile drafting problems and practices. The total number of individuals involved exceeds 118, since several of the subcommittees have found it necessary to organize subgroups including persons not active on the subcommittee. This is particularly true of the subcommittee on bodies.

Any subgroup responsible for the developments of a particular section is composed of specialists well informed in the particular subject. A program for the development of the section is outlined. The industry is then canvassed for suggestions including details of practice. The returns of the canvass are tabulated on comparison charts. An effort is made at meetings to reconcile conflicts and secure agreement on the best practice. Additional canvassing follows after urging that the best practice for the whole industry be kept in mind. When general agreement has been obtained, the report is edited and submitted to the parent group. In many cases this parent group proposes revisions, some of which may be required to meet the requirements of other subgroups. These revisions may require additional study and canvassing, and in many cases joint discussions between subgroup and parent committee. This work is tedious, but the nature of the problem compels such detailed attention.

The SAE Drafting Standards Committee has followed the usual SAE procedure in taking the following steps:

1—Assemble men with the necessary information and experience on all committees.

2—Collect and examine all of the pertinent facts.

3—Develop a general understanding of all of the problems and issues by adequate discussion.

4—Record the best judgment of the group in a recommendation in which every effort is made to approach unanimity.

A few of these subcommittees have preliminary drafts of their reports already under consideration by the editorial committee. It is hoped that this important and difficult project can be substantially completed during 1949.

Advice of ASEE Desired

The automobile industry recognizes the important part that the American Society for Engineering Education will take in any program to produce a drafting manual usable throughout American manufacturing industries. It hopes that the ASEE Division on Engineering Drawing will be willing to give the SAE Automotive Drafting Standards Committee the benefit of its counsel on the results of the latter committee's work. The members of the Division on Engineering Drawing have been giving careful attention to the requirements of many industries which employ engineering graduates and can give automotive men much needed and much appreciated advice on questions requiring careful consideration before the final decisions.

The Division of Engineering Drawing recognizes the importance of any project for standardizing drafting practice. Whereas standards in the mechanical area are subject to frequent change, satisfactory drafting standards can and should have a much higher degree of premanency. Any standard proposed with any authority will undoubtedly become the basis of engineering college instruction. It is important for the student, for the college, and for industry that the young graduate shall find no difficulty in dealing with the drawings of the organization employing him at some later date.

It is recognized that the problem is more difficult in the case of drafting standards than it would be for a mechanical standard. The latter might be justified in case only a considerable percentage of manufacturers would decide to use it. A drafting standard must meet the needs of enough manufacturers so that the percentage of uniformity throughout all industry is substantially increased by its introduction.

Scope of SAE Drafting Standards

The SAE Automotive Drafting Standards will cover general practice and procedure for preparation of engineering drawings. They will not constitute a textbook of instructions on the fundamentals of drafting, in which it is assumed that

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personnel activity engaged in such work will be sufficiently trained. These standards should be found to be useful supplements to the textbooks in use in engineering college instruction in drafting.

At present 14 sections are under preparation, each to be the product of a subcommittee. The fifteenth or Editorial Subcommittee has the task of securing consistency between the different sections. The addition of other sections has been discussed, and may occur.

The present sections are: (1) Drawing Size and Format; (2) Lines, Lettering and Sectioning; (3) Projection; (4) Dimensioning; (5) Abbreviations, Definitions, Notes and Symbols; (6) Threads; (7) Gears, Splines and Serrations; (8) Forgings; (9) Castings; (10) Stampings; (11) Surface Finish; (12) Springs; (13) Bodies; (14) Chassis Frames.

Introduction of SAE Drafting Standards

SAE Drafting Standards will be published and distributed without any commitment as to their finality. It is probable that they will be changed later on to assist in the unification of national and international drafting practices. They will, nevertheless, represent the collective opinion of the automotive industry as to the practice best suited to its own requirements, as influenced by the industry's estimate of requirements of industry at large in any future consolidation or unification of practice. They will not be representative in all details of current automotive practice, but it is hoped that they will tend to consolidate that practice. We can expect drafting standards, after these are published, to be constantly in the hands of draftsmen and their supervisors, and therefore to influence decisions in the same way that standards in the SAE Handbook tend to unify practice.

It should be noted that the Automotive Drafting Standards, when completed, will not be the only SAE effort in this field. The SAE has done standardizing work in aeronautical materials—and in the powerplant area of aeronautical design. The SAE Special Aircraft Project Subdivision has developed an SAE Aeronautical Drafting Manual which has already appeared in a second revised edition, and revisions leading to a possible third edition are under consideration. As in the case of the Automotive Drafting Manual, the committee responsible for the Aeronautical Drafting Manual has been selected from representatives of the interested industry. Differences are inevitable between the aeronautical and automotive groups, and these differences will require adjustment if any manual for general use by all industry is finally to be a matter of agreement.

Use or non-use of any SAE standard is a matter for the voluntary decision of any possible user. The SAE is not in a position to do more than make a recommendation. The manufacturers do not, and in fact could not, enter into any joint agreement with respect to designs, processes or materials. Therefore the degree of industry acceptance of any eventual Automotive Drafting Standards will be determined by decisions taken separately by the different manufacturers. Prior to these decisions there will be a period of study during which adjustments would be possible to conform to any practices recommended in a manual for universal use.

Conclusion

We are very grateful for the opportunity to present a general outline of SAE activity in automotive drafting standards, which we hope will be of interest to the Division of Engineering Drawing. The chairman of our committee is William A. Siler, chief draftsman of the Delco-Remy Division at Anderson, Ind.; the secretary is R. C. Sackett, New Center Building, Detroit 2, Mich. The SAE will be glad to supply additional information.

In conclusion, I wish once more to emphasize our belief that any drafting standard for universal use throughout industry can be successfully developed only by a group which has the complete picture of all industry requirements before it. Such comprehensive knowledge is only obtainable when each industry develops first the complete picture of its own needs, and then takes the trouble to explain its position fully to the others, at the same time maintaining a sympathetic attitude toward the others' problems. We believe this so thoroughly that we have been making a most serious effort to develop the best possible analysis of our requirements. We hope that this will receive consideration in parallel with similar analyses from other industries.

We believe that any attempt to reduce the amount of study which we consider to be necessary will introduce the risk that many industries will not later follow the proposed new standards in some features. The net result could leave considerable deviation. The percentage of acceptance might possibly be so low that the effort expended will be largely wasted.

We believe it would be most unfortunate if any new general standards are issued which fail to receive very much greater acceptance than have similar efforts in the past.

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If so, please notify the secretary of your local Section, and also SAE Headquarters. You will find secretary addresses in this issue on p. 94

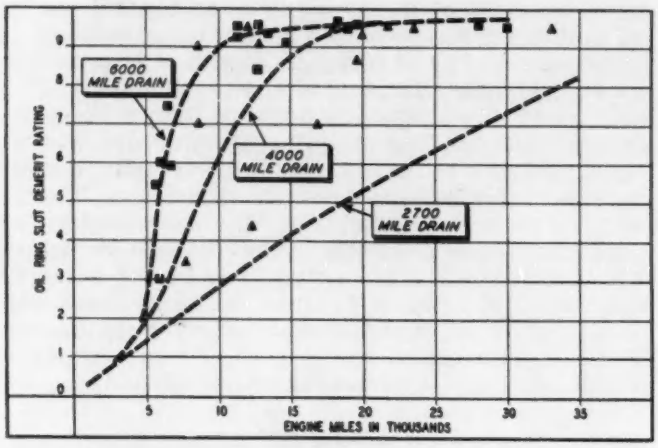

Fig. 1—Effect of oil drain period on oil ring plugging

Why and When TO

ANSWER to the question "Why change oil," is that when oil condition deteriorates (usually due to contamination), engine condition deteriorates.

Answer to the question, "When to change oil," appears to us to be, whenever the oil condition becomes poor.

It is pretty generally recognized that today's high-quality motor oils do not "wear out" or lose their lubricating ability with extended use—except under extremely high-temperature operation. But engine oil does accumulate contamination due to engine malfunction and from the atmosphere. And it is engine and operating conditions that govern proper frequency for draining.

Prominent contaminating elements include unburned fuel, partially burned fuel (commonly known as fuel soot), water from condensation or cooling system leaks, road dirt, lead salts, and to varying degrees products of decomposition of the oil itself. Such contamination manifests itself in engine performance in the form of sludge or varnish deposits on engine working parts which may result in oil screen plugging, oil line clogging, piston-ring-zone deposits (oil-ring clogging and ring sticking), valve sticking, accelerated wear, and lowered engine efficiency.

Field Test Conditions

Behind these conclusions are data obtained during an extensive field test program, under closely supervised conditions, in which the mileage drain interval has been related to actual engine condition. The field test program and laboratory work were carried out by the Esso Laboratories, Research Division, Standard Oil Development Co., except where reference is made to other sources.

Data were obtained from a commercial fleet of 50-odd vehicles working during the night in heavy-duty intercity freight service (average payload 24,000lb) and in local delivery service in the terminal areas during the daytime. New engines were installed in each piece of equipment at the start of the test and each engine was dismantled for rating at specific intervals. A high quality 2-104B engine oil was used in this test.

Demerit Rating System

For comparing engine conditions, we have used an arbitrary demerit rating scale of 1 to 10; 0 representing a perfectly clean part and 10 a completely dirty part or one in which failure would be imminent. In cases where overall engine sludge ratings are shown, the overall rating is a numerical average of the ratings of individual parts.

Table 1 shows the overall sludge demerit after 15,000 and 30,000 miles.

Similar data on the rate of oil-ring clogging (Fig. 1) show that the oil ring holes were completely plugged after 12,000 miles of operation on 6000-mile drains and after 20,000 miles on 4000-mile drains. But oil-ring slots rated less than 7.5 on the demerit scale at 30,000 miles of operation on 2700-mile drain periods. Complete plugging under 2700-mile-drain operation was not anticipated before 40,000 to 45,000 miles.

These data were obtained using a drilled oil ring that has since been replaced with a slotted oil ring by the engine manufacturers. Additional informa-

Table 1—Overall Sludge Demerit

	After 15,000 miles	After 30,000 miles
2700-mile Drain Period	1.4	1.9
4000-mile Drain Period	1.8	2.5
6000-mile Drain Period	2.4	3.0

* Paper "Why Drain Oil?" was presented at SAE Buffalo Section on March 29, 1949. (This paper is available in full from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

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DRAIN OIL

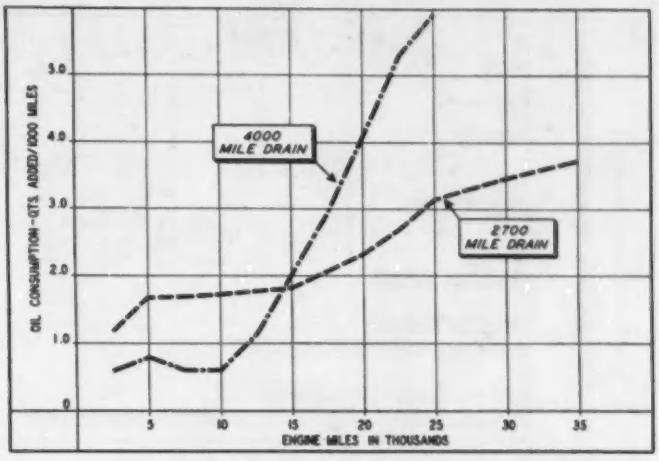

Fig. 2—Effect of oil drain period on oil consumption

EXCERPTED FROM PAPER* BY

W. G. Perriguey Sales Engineering Division
Esso Standard Oil Company

tion indicates that the effect of oil drain interval on rate of oil ring clogging is less pronounced with the newer type ring, but the difference is still appreciable.

Also, preliminary evaluation of cylinder wear data collected during this test indicates that wear rates steadily increase with the longer oil drain periods.

The main advantage put forth for extended oil drain intervals is reduction in overall oil consumption. While the effect of oil ring clogging upon oil consumption varies considerably among engines of different design, in the group of test vehicles reduction in overall oil consumption was not in direct proportion to the extension of oil drain intervals. The cleaner oil ring condition obtained when operating at the shorter drain interval resulted in a marked reduction in make-up oil consumption. Fig. 2 shows the effect of oil drain period on oil consumption.

Hand in hand with plugging of oil rings goes sticking of compression rings. Stuck compression rings lead to inefficient combustion and rapid loss of engine power, as well as further contamination of the lubricating oil, which accelerates the entire cycle.

Even more dramatic than the effects of deposits on rings are the effects of deposits on oil pump screens and in oil pressure lines. When either of these components becomes completely plugged, engine failure is almost instantaneous. Table 2 shows incidence of engine failure or severely impaired operating efficiency caused by engine deposit build-up.

Of course, these tests represent a specific set of operating conditions with a single make and model of engine. The numerical ratings are not necessarily applicable to another type of engine or to another set of operating conditions.

While the operation would normally be identified as heavy-duty service, the deposits formed correlate more closely with laboratory test engine results obtained under the low-temperature operating cycle than they do to the L-4 high-temperature engine test schedule. This leads to the conclusion that the local-delivery operation followed during the daytime with this equipment may be the more important portion of the cycle with reference to engine deposits and the factors that influence choice of proper drain interval.

Fuel used in these tests was an experimental fuel of a composition more severe in deposit-forming tendencies than the fuels generally available for commercial service. However, other work with fuels of varying cleanliness characteristics indicates that even where fuels of exceptional cleanliness characteristics are used a similar relationship exists between drain interval and deposit build-up, although at a lower level of deposits.

To determine the effect of filtration, ventilation, and engine temperatures, further tests were conducted.

Filtration—Some of the engines were fitted with filters. Filter change practice was established in relation to the rate at which filters became loaded. After 12,000 miles, engines equipped with filters showed about 55% less in sludge deposits than non-filter-equipped engines. The big improvement was reduction in deposits on oil screens and in oil lines and in general engine cleanliness. There was little

Table 2—Effect of Drain Interval on Engine Failure Resulting from Deposit Formation

Cause of Failure on Inefficient Operation	% of Test Unit Failures for Oil Drain Period		
	2700 miles	4000 miles	6000 miles
Oil Screen plugging (>30%)	8	14	40
Oil Line Plugging (Complete)	8	28	60
Oil Ring Plugging (90% at 20,000 miles)	0	100	100
Average Engine Life, miles	38,500	30,850	16,890

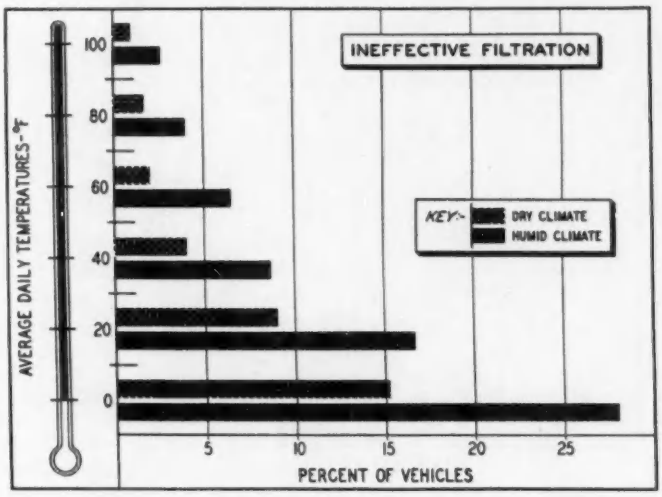

Fig. 3—Vehicles showing ineffective filtration or loaded oil filters as function of average daily temperature

or no reduction in varnish or ring-zone deposits.

This fact may indicate that only the sludge and small parts of the varnish deposits are precipitated from the circulating oil, while the other deposits are formed directly on the engine parts without first passing into the oil.

Crankcase Ventilation—Laboratory tests were conducted using basic Chevrolet FL-2 test procedures but varying the amount of crankcase ventilating air from the standard value of 60 cu ft per hr down to zero. Pertinent results indicate that valve deposits increased by 60% and sludge deposits increased nearly 400% with no crankcase ventilation.

Engine Temperatures—Results of a similar test using the basic FL-2 schedule with several combinations of jacket and crankcase temperatures showed that the minimum sludge and varnish formation occurred at high jacket temperature (175 to 180 F) when crankcase temperature was intermediate (180 to 230 F). On the other hand, ring-zone deposits were minimum at lower jacket temperatures (130 F) with intermediate crankcase temperatures (about 190 to 230 F). Consequently, when conditions are such that jacket or crankcase temperatures depart from the optimum, oil and engine conditions deteriorate at a more rapid rate and the serviceable life of the oil is therefore shortened.

Effect of Climate Variations

The degree to which variations in climatic condition influence rate of contamination in the average vehicle engine is illustrated by data obtained by the U. S. Army Ordnance Department during the war. Analysis of the data showed that as average daily temperature went down, the rate of oil contamination markedly increased. Also, for any average daily temperature, the rate at which filters loaded was almost twice as great in humid areas as it was in dry areas (Fig. 3). In other words, the colder and damper the climate, the more rapidly the oil becomes contaminated. Consequently, it is desirable to drain the oil more frequently.

Also, if operating conditions cause low engine temperatures, even though the atmospheric temperatures may be relatively warm—such as results from intermittent stop and start driving—the drain interval should be correspondingly reduced.

To study effect of passenger-car drain intervals, a parking lot survey was conducted in 1944. Among units driven 25,000 to 40,000 miles, those operated with oil drain periods of 1500 miles or less showed an average sludge demerit of only 1.1. Those operated with drain periods over 1500 miles averaged 2.7. Cars having gone 25,000 to 40,000 miles with moderate drain periods were as clean as the average of all cars having operated 10,000 to 25,000 miles. Thus, from a deposit standpoint, moderate drain periods had appreciably postponed the advent of middle age in these engines.

Cars with oil filters had less sludge than cars without oil filters, provided that filter maintenance was reasonably good. However, where filter maintenance was poor or the filter was never changed, the sludge accumulation was even greater than where no oil filter was in use. Specifically, the average sludge demerit with good filter maintenance was 1.2, with no filter was 3.1, and with poor filter maintenance was at the dangerous level of 4.1.

When to Change Oil

These test results indicate that oil should be changed because engine condition deteriorates with oil condition and that the time to change oil is whenever oil condition becomes poor.

But they do not disclose how a vehicle operator is to know when the oil condition becomes poor.

The answer to this question, as we see it, differs for the fleet operator and the operator of an individual passenger car.

In setting drain periods, fleet operators should consider the effects of three factors upon rate of crankcase contamination:

1. Climatic conditions
2. Type of operation (long-haul heavy-duty service or short-haul intermittent service)
3. Maintenance of combustion efficiency, engine temperatures, ventilation, and control of atmospheric contamination

Used-oil analysis can be very helpful also in arriving at the most practical drain interval for fleets.

If the passenger-car operator has facilities available for controlling maintenance on his car on a level comparable to that available to most fleet operators, there is no reason why the extended drain practices followed by many fleets can not also be followed with satisfaction for the passenger car. Also, if he wants to take the time and trouble to acquaint himself with the various climatic and operational factors which influence rate of contamination, the passenger car owner can modify his drain practice in accordance with the probable requirements of his particular car, thus obtaining some moderate economies in his lubricating oil expense.

But, for most passenger car operators, the solution to the problem of how often to drain oil is to follow conservative drain intervals, since the cost of oil is nominal in comparison to the cost in reduced engine life or performance that may result from over-extended drain intervals.

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Selection of Steel for Automobile Parts

What Engineers Should Know Today About Hardenability-Band Steels

Part III—Steel Composition Related to Hardenability

This is the third of a six-part report issued by the SAE Iron & Steel Technical Committee that is appearing serially in succeeding issues of the SAE Journal. The series started in the August issue. This report was prepared at the request of the SAE Iron & Steel Technical Committee's Division XVIII, Hardenability Publications. Part I was prepared by Joseph Geschelin, Chilton Co., from material provided by the Committee's Division III, Hardenability Bands. Parts II-VI were prepared for the Division by A. L. Boegehold, Research Laboratories Division, GMC.

NOW that we have had a brief outline of hardness and strength requirements, we will proceed to discuss how we go about determining the composition of the steel which will produce the desired properties.

In the days when the hardenability of steels was not known, metallurgical handbooks and metallurgical literature were copiously supplied with physical property charts, like the one shown in Fig. 5. This shows how hardness and tensile properties change with increasing tempering temperature after hardening. These charts were presumably to assist in selecting steel for various applications; but one looks in vain for significant differences between curves of this kind for a number of different types of steel having similar carbon content.

All charts are very much alike,

Composition, %		*Approx. Critical Range*		*Heat Treatment*
Carbon	0.35-0.45	Ac_1	1355° F.	Quenched from 1475°-1525° F.
Manganese	0.60-0.90			
Phosphorus	0.04 Max.	Ac_2	1380° F.	Quenched into Oil
Sulphur	0.05 Max.			
Nickel	1.00-1.50	Ac_3	1395° F.	Tempered as Indicated
Chromium	0.45-0.75			

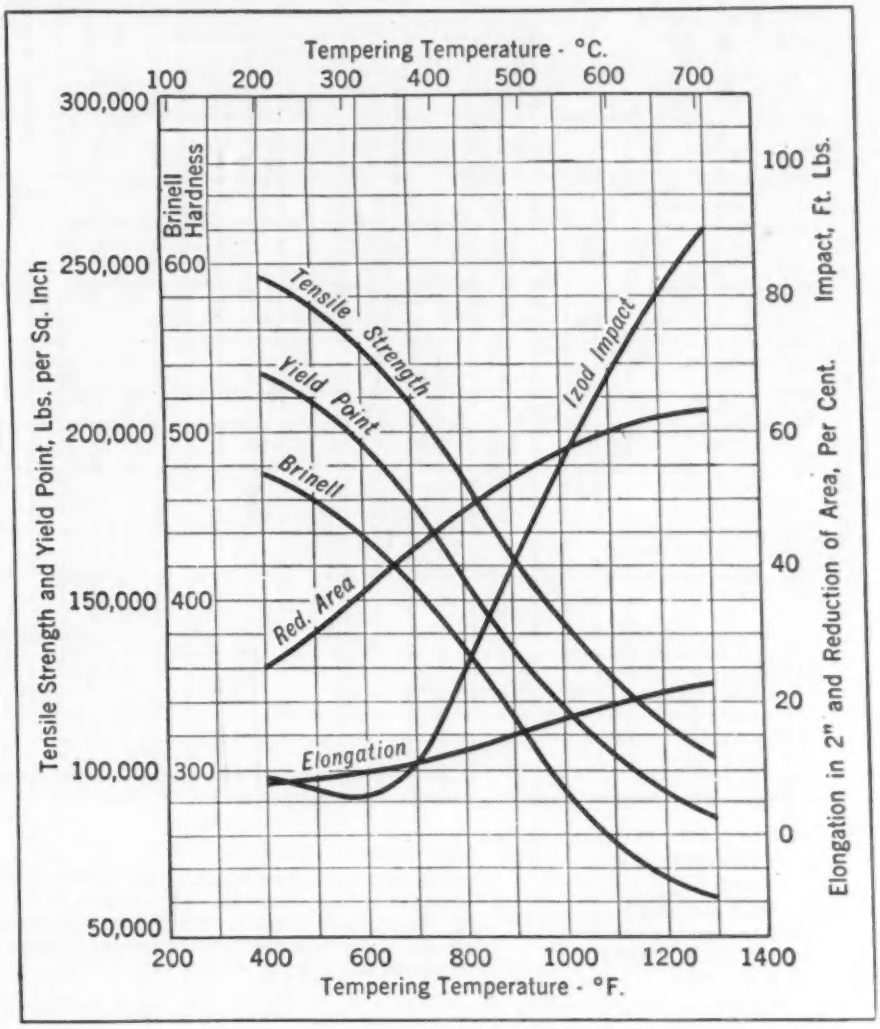

Fig. 5—Mechanical property chart for nickel-chromium steel, SAE 3140, oil quenched

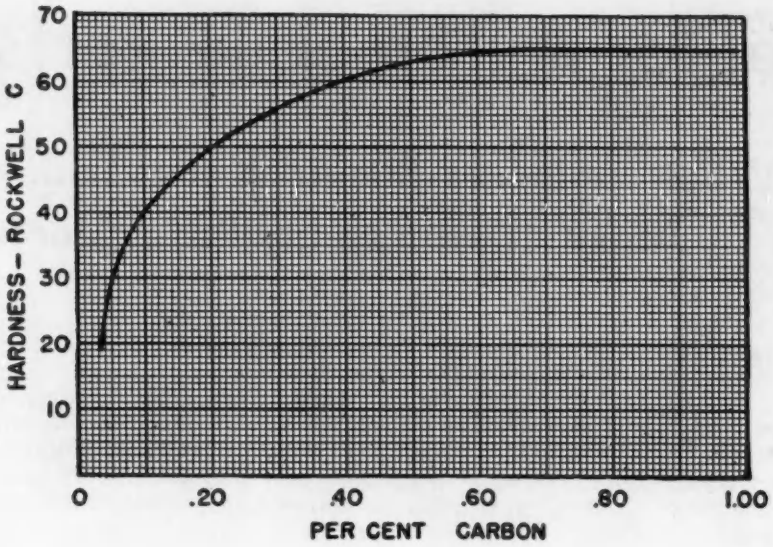

Fig. 6—Maximum hardness obtainable in plain carbon or low alloy steel, according to article by Burns, Moore, and Archer in American Society for Metals Transactions, 1938, Vol. 26, p. 14

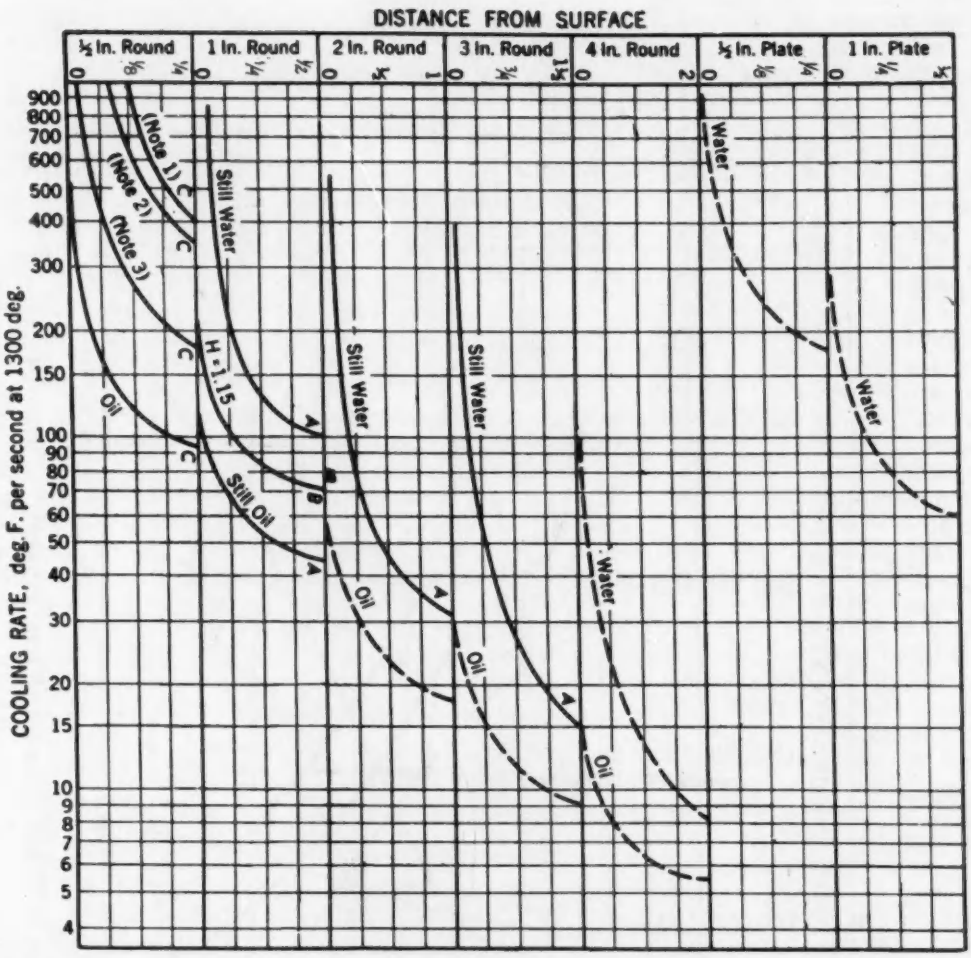

Fig. 7—Cooling rates in commonly used section sizes

Note 1 - 5 per cent NaOH at 70 deg. F.

Note 2 - Water at 70 deg. F.

Note 3 - 5 per cent NaOH at 175 deg. F., or water at 115 deg. F.

References-

A - Howard Scott - Trans. A.S.M. Vol. 22, 1934, P. 68-94.

B - Asimow, Craig, Grossman - S.A.E. Journal; Vol. 49 1941 No. 1

C - French, Hamil - Trans. A.S.S.T. Vol. XVI, 1929, P. 711-739.

Center Cooling Rates of Estimated Curves from French. The Quenching of Steels. A.S.M. 1930, and Scott.

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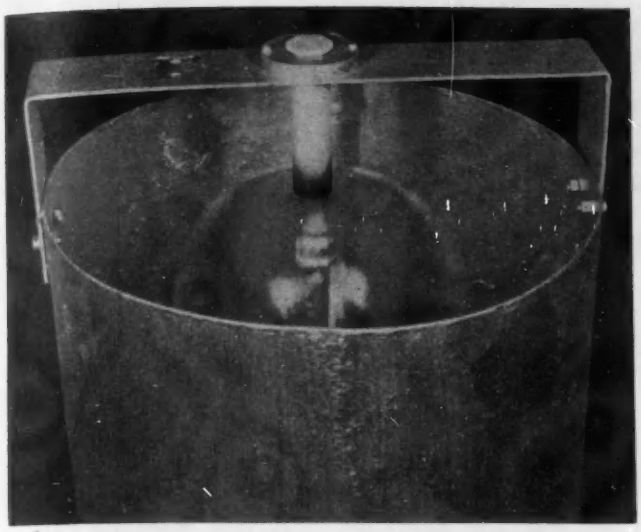


Fig. 8—Jominy hardenability bar being quenched on the end

the greater is the difference between surface and center cooling rates.

The total range of cooling rates that occurs anywhere in the size bars shown is from 4°F per sec up to 600°F per sec. In order to study all kinds of alloy steels throughout this range of cooling rates, we developed a single test specimen containing this entire range of cooling rates. This is called the Jominy hardenability bar, or the end-quenched test bar. The steel to be studied is made in the form of this test bar, heated for hardening and then quenched on the end, as shown in Fig. 8, thus cooling at different speeds over the entire length, as shown in the chart in Fig. 9.

Hardness tests are made along the side of the test bar and plotted as in Fig. 10, thus giving us the hardness that can be obtained at all of the cooling rates to be found in various size sections used in automobile parts. The hardness at the 1/16 point will be determined by the carbon content and as we shall see later, the distance before hardness drops off depends on the alloy content.

so much so that it is possible to dispense with charts of this kind for design work in favor of charts which show the relationship between hardness and the other tensile properties, as applied to all steels, for that is all that can be learned from such a chart. We have to have information on hardening power to decide which steel to select for a given part.

To begin with, the carbon content of the steel must be high enough to develop the desired hardness. The hardness obtainable depends on the carbon content in the manner shown in Fig. 6. The quenching conditions necessary for obtaining the hardness for each carbon content shown in Fig. 6, however, is not given. The fact is that, with nothing more than carbon in the steel, the rate of cooling from the hardening temperature to produce hardening would have to be exceedingly fast. The rate at which a piece of steel can be cooled in the quenching operation depends on the size and shape of the piece and on the quenching medium.

Therefore, to obtain a cooling rate fast enough to obtain the maximum hardness in a steel with only carbon in it, a very small piece of it would have to be quenched in water or brine. Later on a chart will show data on this subject. But as a single example, steel containing 0.40% carbon with no alloy in addition, would be half hard or 42 Rockwell C at the center of a 0.213-in. diameter round when quenched as fast as possible in high-velocity caustic soda solution.

The cooling speed required to harden fully a certain steel is called the critical cooling speed. Whether a piece of steel will harden fully or not is determined by whether it can be cooled faster than the critical cooling speed of the steel from which it is made. The critical cooling speed of the steel is a function of the carbon and alloy content of the steel.

Fig. 7 shows the cooling speeds that occur in various size sections common to automobile parts. One fact that may be observed is that the surface of any size piece cools a great deal faster than the center, and that the more severe the quenching method,

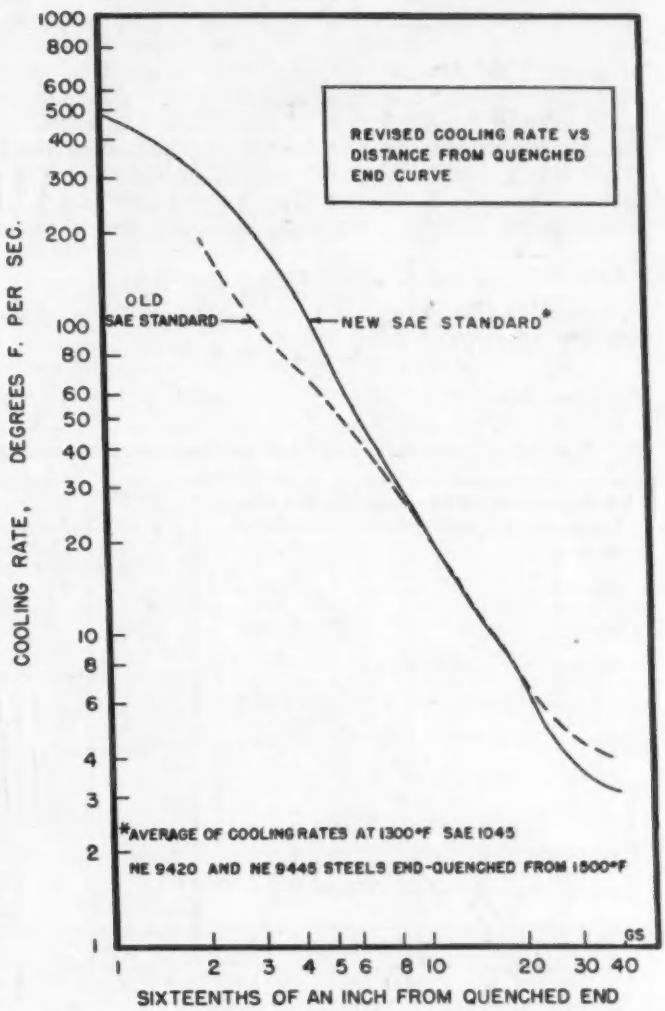


Fig. 9—Cooling rates over the entire length of the end-quenched test bar

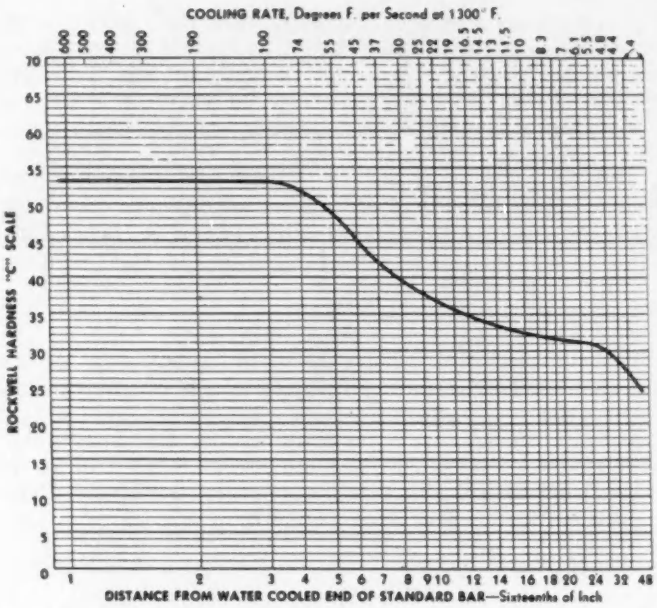

COOLING RATE, Degrees F. per Second at 1300° F.
ROCKWELL HARDNESS "C" SCALE
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

Fig. 10—Hardenability of SAE 5130 heated to 1675F and end quenched

With the creation of this test method, metallurgists were provided a convenient means of determining many facts about steel which had gone undiscovered because of the large amount of work necessary to dig out the information. One of the first uses of the test was to determine the effect of alloying elements on the hardening power of steel. This was done first by Grossman of Carnegie-Illinois Steel Co. He found that each alloy added to steel has a certain effect of reducing the cooling speed required to harden the steel. The amount this cooling speed is reduced varies with equal additions of different alloying elements.

The relative hardening power of carbon, manganese, chromium, nickel, and molybdenum are shown in Fig. 11[1]. The chart at the left shows the effect of carbon content by itself. The numbers along the vertical axis represent the diameter bar which, when quenched with an "ideal" quench, will harden to 50% martensite at the center. An "ideal" quench is a nonexistent one which will remove heat as fast as it is conducted to the surface, and which cools the surface instantaneously to the temperature of the quenching medium. The chart at the right gives multiplying "numbers" which are used to multiply with the diameters found in the left-hand chart.

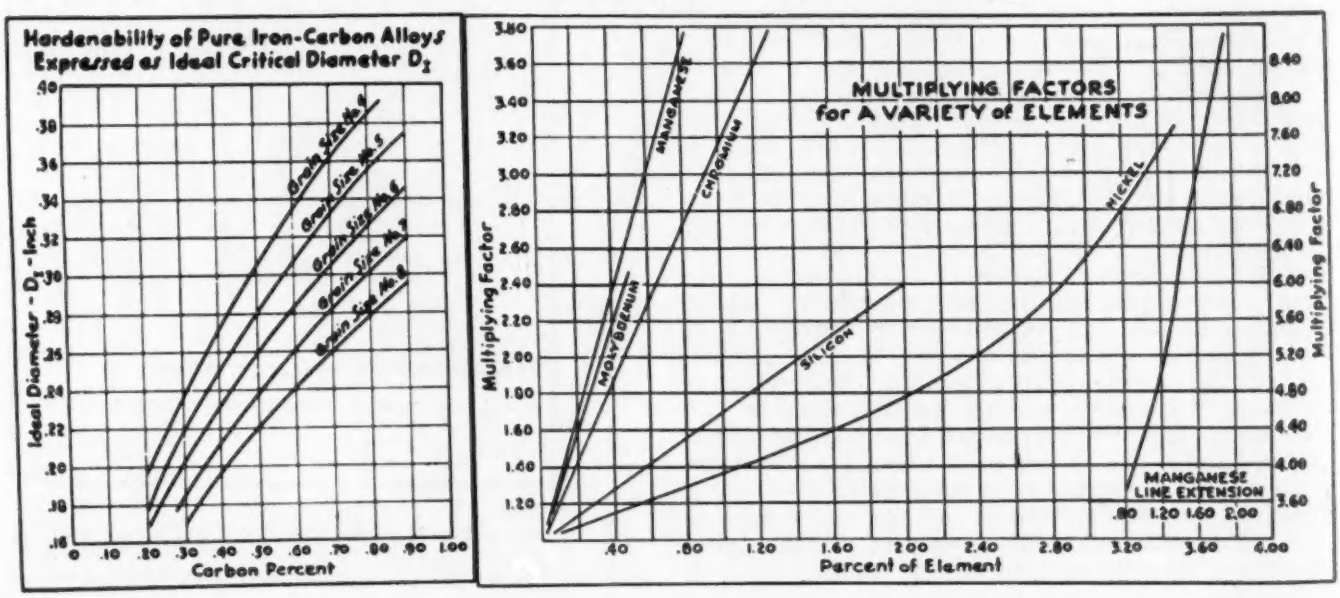
By using the multiplying numbers for the several alloys in a certain steel, we get a product which is the ideal critical diameter for that steel. Knowing the carbon content and the ideal critical diameter, we can then construct the Jominy hardenability curve for that steel by a method described by J. Field[2]. Timken, Bethlehem, and Carnegie-Illinois Steel Companies have constructed slide rules for calculating Jominy hardenability curves from chemical composition using the principles explained by Grossman and Field. (See Fig. 12.) It will be seen that the amount and kind of alloying elements in steel will determine the shape of the Jominy hardenability curve.

As the chemistry of a given specification steel varies from heat to heat within the limits allowed for the different elements, the hardenability curve will vary. Just how much variation in hardenability would occur within the chemical limits for each alloy steel specification was the answer sought several years ago during the war by a committee sponsored jointly by SAE and the American Iron & Steel Institute. Hardenability curves on hundreds of heats of each steel were collected and studied. The spread of hardenability calculated from chem-

[1] AISI—"Contributions to the Metallurgy of Steel—No. 12".

[2] J. Field—Calculation of End Quench Hardenability from Chemical Composition. Metal Progress, March 1943.

Fig. 11—Charts showing the relative hardening power of carbon, manganese, chromium, nickel, and molybdenum



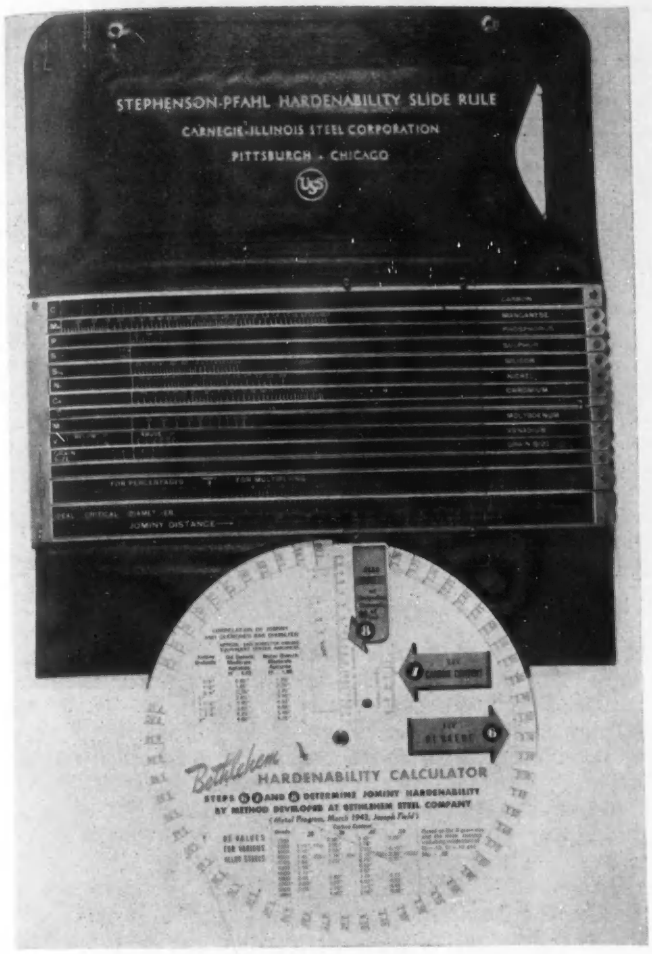


Fig. 12—Slide rules for calculating hardenability curves from chemical composition

istry indicated that the variation in hardenability for any steel would be great; but the spread shown by actual hardenability tests was even wider than expected, due to the influence of residual alloying elements in some heats. It seemed obvious that steps should be taken to protect the steel user by restricting hardenability within narrower limits, so a plan was evolved for accomplishing this.

The result is that hardenability limits have been set up and made available as standard specifications for 62 different alloy steels. These standards have been published by the SAE and AISI jointly in "Contributions to the Metallurgy of Steel—No. 11," under the title "Hardenability of Alloy Steels," dated June, 1947.

The difference between the new hardenability limits and the spread of hardenability incurred by specifying according to chemistry is illustrated in Fig. 13. Two dotted hardenability curves show the spread of hardenability encountered in a large number of heats of 8640 ordered to chemistry limits, and two solid hardenability curves show the limits of hardenability guaranteed by the steel maker if 8640H is ordered—according to the hardenability limits.

The method and statistics on several hundred heats of each type of steel used in deciding on the location of the limits for the new H bands has been described by Mitchell[3].

Over a period of years the steelmakers found that in making steel to chemistry specifications, about 7% of the heats made failed to meet the specifications, and had to be diverted to other uses. It was decided, therefore, to discard 7% of the heats in-

[3] John Mitchell—Progress report on Hardenability Bands, 1945 Yearbook of American Iron & Steel Institute.

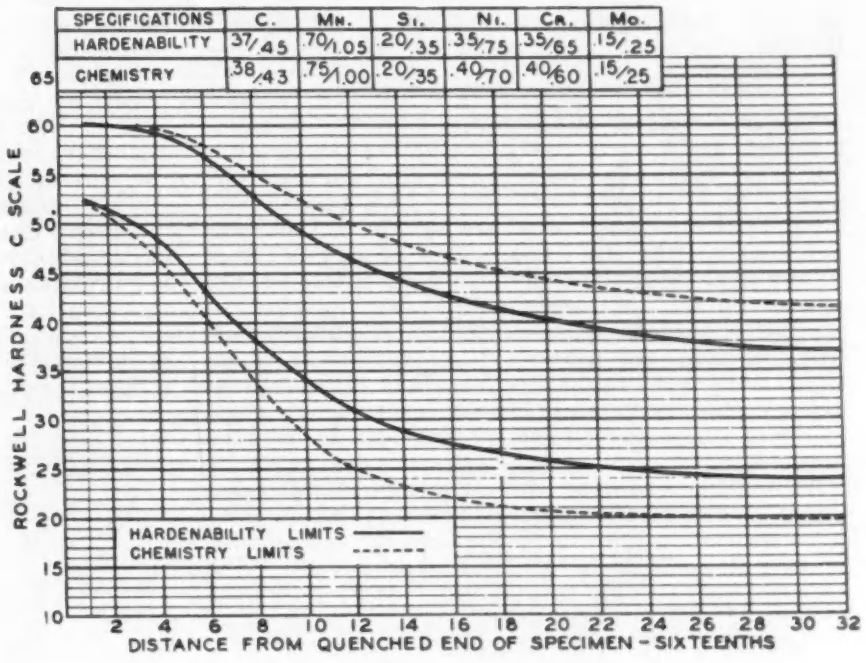


Fig. 13—Hardenability limits of 8640 steel ordered to H-bands (8640H) compared to those by specifying according to chemistry

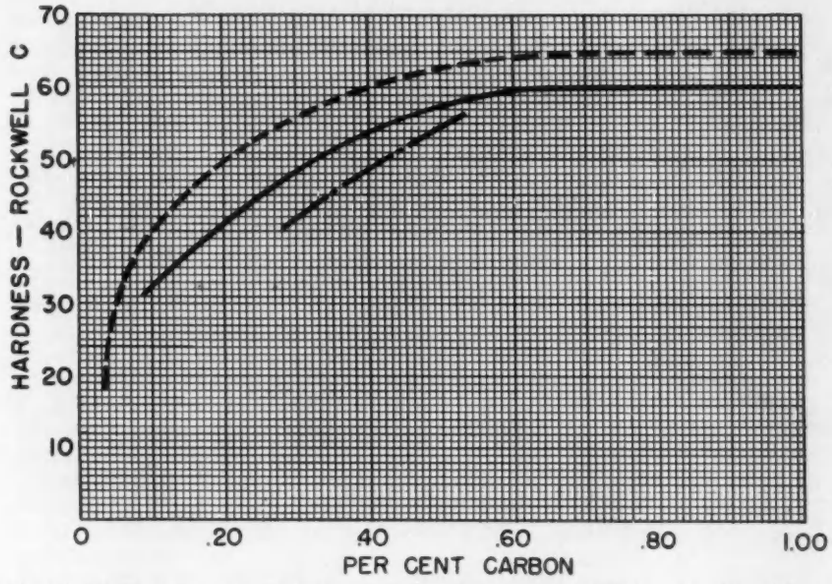


Fig. 14—Minimum hardness at 1/16 in. from quenched end of Jominy hardenability test bar. Dotted line represents maximum hardness obtainable; solid line, minimum at 1/16 in. on hardenability bar for alloy steels; dash-dot line, minimum at 1/16 in. on hardenability bar for carbon steels. (These curves represent minimum values for 90% of the 1400 heats studied. Values for 10% of the heats fall below these curves)

stead, having hardenability on the high or low side. This resulted in the remaining heats having a substantially narrower spread of hardenability. The only difference is that the heats that fall outside the H-band limits and have to be diverted are not the same heats that fall outside the chemistry specifications.

Since the number of rejected or diverted heats involved is no greater than formerly, the customer can specify the narrower H-band limits without paying any extra. Notice that the chemistry limits allowed for the H-band specification are wider than the limits when ordering to chemistry alone. This does not mean that all elements could be at low limits or high limits in one heat of steel ordered to H limits. That would result in hardenability outside the band limits. It means that chemistry can be anywhere within the limits shown, if at the same time the hardenability curve is inside the H-band limits.

H-Band Specs More Logical

The way it works out is that most of the chemical elements are toward the middle of the range, while one may be near the high or low limit for that element. Steel ordered to chemistry limits could have all elements but one toward the middle of the range and one outside the limit. The hardenability of such a heat would be well within the H-band range and would be a perfectly satisfactory steel for the purpose intended; yet because of one element outside the limit, the heat would be subject to rejection. We are talking, of course, about normal times when metallurgists were able to exercise the right to reject when steel chemistry was outside specified limits.

As another example, the chemistry specification would permit all the chemical elements in one heat to be on the low limit or on the high limit, thus presenting a problem involving too low or too high hardenability. This cannot happen when steel is ordered by the H specification. On this basis there is really no justification for ordering steel to the old chemistry limits.

Some metallurgists have claimed that by ordering to chemistry, hardness variation at the surface of quenched pieces is less because of the narrower range of carbon which controls hardness. This is a fallacy. The carbon range of the chemistry specification applied to more than one heat is 9 points, when 2 points allowance for check analysis above and below the 5-point range is included. The total range for the H specification on this basis is 12 points. The fact is, however, that the steelmaker may not allow that much variation in carbon, for if he did, the hardenability curve at the 1/16-in. point would not be within the prescribed limits for hardness which are no wider than the spread encountered when the chemistry specification is used.

Wider Hardness Limits with Chemistry

As a matter of fact, to illustrate with 8640, the steel whose H-band is shown in Fig. 13, the actual hardness range, corresponding to the carbon range permitted by the chemical specification 0.36 to 0.45, would be 50 to 62 Rockwell C. (See Fig. 14 from paper by Boegehold, SAE Journal, Vol. 52, Oct. 1944.) We see, therefore, that ordering steel to chemistry will result in more hardness spread at the surface of hardened articles than would occur when the steel is ordered to the hardenability limits.

As to a chemistry specification plus a minimum hardenability, it should be noted that no advantage is gained that cannot be obtained with the H-band specification. The chemistry plus minimum hardenability method sacrifices control on the high side of the band which often results in cracking and distortion trouble.

Copies of the complete six-part series on Hardenability (SP-59) are available from Special Publications Department, Society of Automotive Engineers, 29 West 39th Street, New York 18, N. Y. Price: $1.25 per copy to SAE members, $2.50 to nonmembers. Quantity prices on request.

Oldsmobile's ROCKET Engine

EXCERPTS FROM PAPER* BY

J. F. Wolfram

Chief Engineer,
Oldsmobile Division, GMC

THE Oldsmobile Rocket engine design is based on these five principles:

1. Design the engine structurally so that it will operate smoothly under the high pressures of 12 to 1 compression ratio. This means stiffness in the crankline never before contemplated in an automobile engine.

2. Overhead valves are the best way to get proper breathing of the engine with the small combustion chamber required by compression ratios above 8 to 1. In an L-head, the small combustion chamber restricts the flow of gasoline vapor from the valves to the piston.

3. Design the engine for cooler intake manifold temperatures and adequate valve cooling to reduce engine detonation and preignition.

4. Design the engine of such size that it will give the desired performance with a compression ratio suitable for fuels available at the time the engine is put into production.

5. Design the engine so that it can be increased from 7.25 to 1 compression ratio to 12 to 1 with a minimum tooling change.

Discussed below are some of the Rocket engine's design features and performance characteristics stemming from adherance to these principles.

* Paper "Oldsmobile Rocket Engine," was presented at SAE Detroit Section, Lansing, April 5, 1949. (This paper is available in full in photolithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

1. General Design and Specs

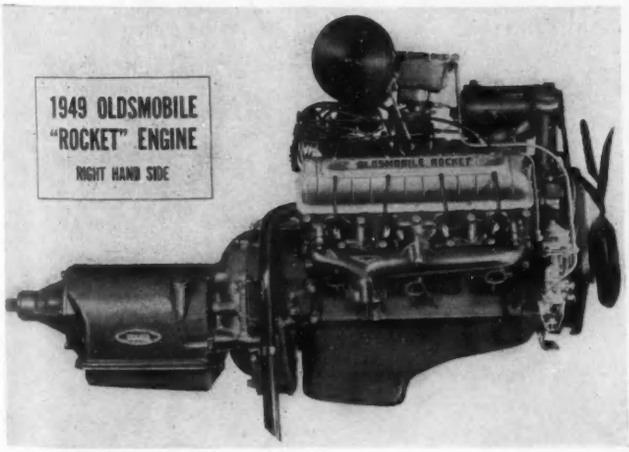


Table I—General Specifications of the Oldsmobile "Rocket" Engine

Displacement	1948 Line-8 257	1949 "Rocket" 303
Bore	3 1/4	3 3/4
Stroke	3 7/8	3 7/16
Max. Installed Hp	99	122
Max. Installed Torque	194	240
Stroke/Bore Ratio	1.19	0.92
Weight Per Hp	6.2	5.5
Compression Ratio	6.5 to 1	7.25 to 1

Two factors dictated the choice of a 90-deg V-8 engine. First, the V-8 design with five main bearings offers a comparatively economical means of getting the necessary structural stiffness in the crankline. Second, this design—the block length

of which largely is determined by the space requirements to accommodate five main bearings—provides space for large diameter bores. This allows a large bore-stroke relationship, which results in improved mechanical efficiency, as the specifications above indicate.

In the general view of the engine above, showing location of accessories, note that the air cleaner and silencer assembly is located cross-wise of the engine behind the side-entrance carburetor. The distributor is mounted at the rear below the air cleaner, and the ignition coil is a few inches in front of it.

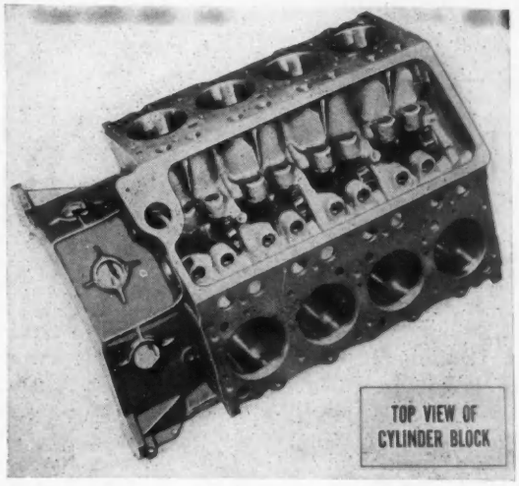
The generator is at the front, above the intake manifold. It is driven from the fan pulley by a separate belt of narrow wedge design. The cast front engine cover houses the single impeller water pump and contains the water inlet and outlet passages. The fan, mounted on the front of the water pump, is driven at 80% engine speed for minimum noise.

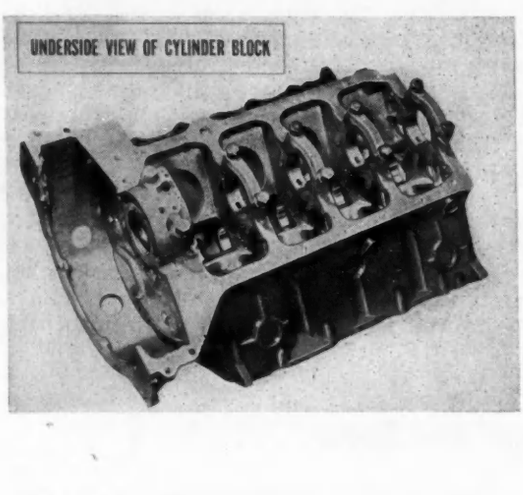
The front motor mount is bolted to the bottom of the cast cover; the oil filler tube and breather are located at the top; and the fuel and vacuum pump on the side. This pump is driven by an eccentric bolted on the end of the camshaft and the vacuum booster part of this pump is double-acting to give extra capacity to drive the large wiper on the larger 1949 windshields.

2. Cylinder Block

Note in the underside view that the cylinder block, crankcase, and upper half of the flywheel housing are a single casting. The lower surface of the cylinder block is approximately on the crankshaft centerline. This block design makes for a short engine of exceptional rigidity.

The top view shows some of the casting details of the cylinder block upper compartment. Early in our experimental program we found that if openings were left in the "V" of the block below, the lifter bosses, oil thrown off the drive line was splashed on the inner water jackets. This produced a top oil temperature, under severe operating conditions, about 25 deg cooler than in the 1948 Line-8 engine.


TOP VIEW OF CYLINDER BLOCK

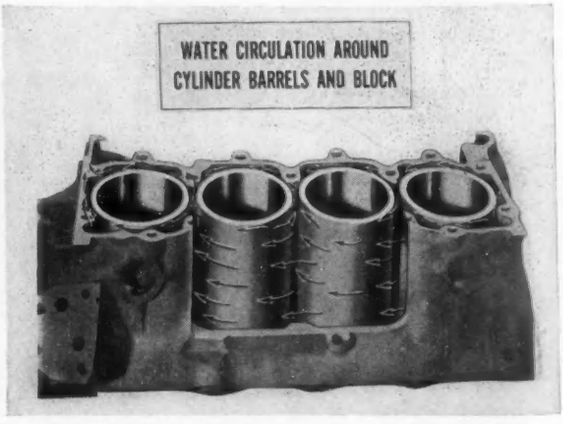

UNDERSIDE VIEW OF CYLINDER BLOCK

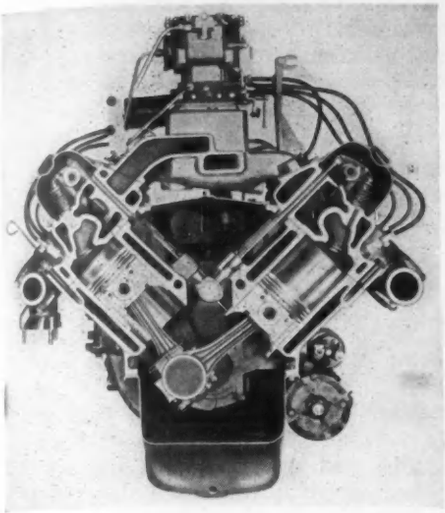
Additionally, the warm-up rate of the oil on a cold start was found to be about twice as fast as in the Line-8, a desirable feature in retarding sludge formation.

3. Water Jacket Design

Shown at right are some of the details of the block design in the water jacket and around the cylinder barrels. The cylinders are completely water-jacketed. The cylinder head bolt bosses are tied into the side walls of the water jacket and the side walls are scalloped to provide maximum support for the bosses. In this way the bolt bosses are divorced from the cylinder walls, minimizing cylinder distortion when the head bolts are tightened.

The scallops in the water jacket walls also help in directing the water around the cylinders. There is a minimum water space of 5/16 in. between barrel walls, which allows unrestricted flow.


WATER CIRCULATION AROUND CYLINDER BARRELS AND BLOCK



4. Internal Arrangement

The transverse section through the center of the engine, at left, gives the general arrangement of internal parts. The camshaft is at the center of the "V." It operates the valves through hydraulic lifters, push rods, and rocker arms. Oil to the upper valve mechanism is metered through holes in the camshaft bearing journals.

The combustion chamber incorporates a quench area for detonation control. The combustion chambers are "as cast" and held to limits of $\pm$ 1 cc.

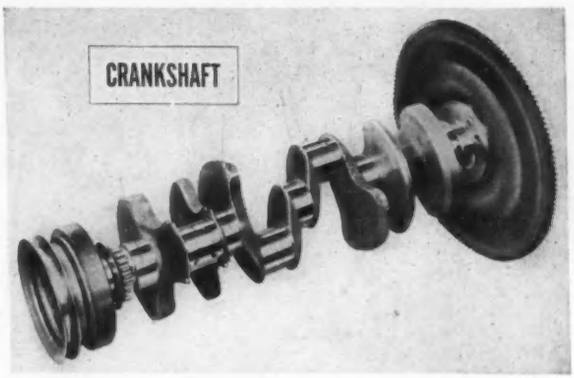
The cross-section shows the down draft induction system which results in excellent distribution and reliable cold starting, even at sub-zero temperatures. The exhaust manifold arrangement on the outside of the head results in short exhaust passages which reduce heat rejection to the water.

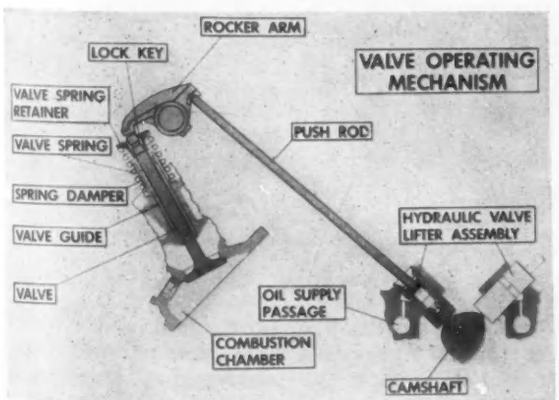
5. Crankshaft

The crankshaft is supported by five main bearings. Its overall length is 25 7/16 in., as compared with 39 3/16 in. for the 1948 Line-8 shaft. The centerline distance between main bearings is 4 5/8 in. as compared with 8 1/16 in. The average overlap between connecting rod and main bearing journals on the Rocket engine is 11/16 in. compared with 13/32 in.

Crankshaft rigidity was found to be a major requirement for engine smoothness in high compression engines. Oldsmobile worked with GMC Research Laboratories on a procedure for static evaluation of crankshaft stiffness. This involves supporting a section of a crankshaft on the rear two main bearing journals and applying a 1000-lb. load to the crank pin journal between.

Under these conditions, the angular deflection of the flywheel flange is measured. The Rocket engine crankshaft on this test showed a deflection only one-third as great as the Line-8 shaft. The crankshaft is very rigid torsionally. Maximum total torsional amplitude we have been able to measure without a balancer is 0.3 deg.





6. Valve Arrangement

The illustration at left shows the arrangement of the valve operating mechanism.

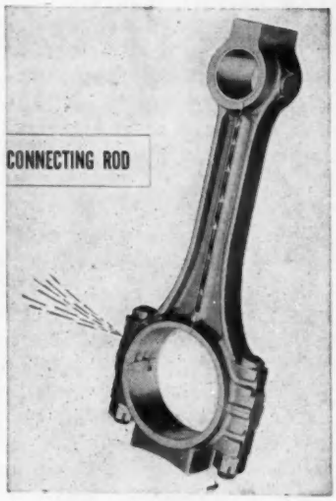
The valves are set at an angle to improve mixture flow and to get a short rocker arm. The exhaust valve stems are 1/16 in. larger than the intakes to improve cooling. They are also undercut at the lower end for improved resistance to sticking.

Exhaust valve material is XCR and the intake is SAE 3140. Oil is fed to the oil reservoir of the hydraulic lifter from an oil gallery in the block.

The lifter has a total automatic adjustment range of 0.170 in. and lifter bodies and plungers are selected to extremely close tolerances to control es-

cape of oil from the pressure chamber at the bottom of the lifter assembly.

Material of the lifter body is high alloy cast iron, heat-treated to give a hardness of from 58 to 61 Rockwell C. This material gave a considerable increase in life over chilled cast iron. The lifter has a 30-40-in. radius to run on 6′-8′ taper cam with 1/16 in. offset.



7. Connecting Rods

The connecting rods are "I" sections with a straight instead of a tapering channel. They have a boss in the center for full oil pressure lubrication to the pin. As shown in the photograph, there is a weight controlling pad on each end of the rod which is machined on an automatic balancing machine in production to $\pm$ 1/16 oz.

The connecting rod bearings are 2 1/4 in. in diameter, 7/8 in. wide, and are steel backed with Durex matrix and babbitt overlay.

A cylinder wall oiling spit hole at the split line indexes with the crankshaft journal oil hole in such a way as to direct oil on the upper side of the cylinders in the opposite bank at low speeds. The oil holes in the journal are offset to introduce the oil ahead of the maximum load position to increase bearing life.

8. Pistons

The Rocket engine uses a three-ring aluminum piston of the steel strut type, shown at right, which has a full floating wrist pin. The piston skirt is cut away at the front and rear of the crankshaft counterweight clearance. In this way a short connecting rod is used to get a minmum block height.

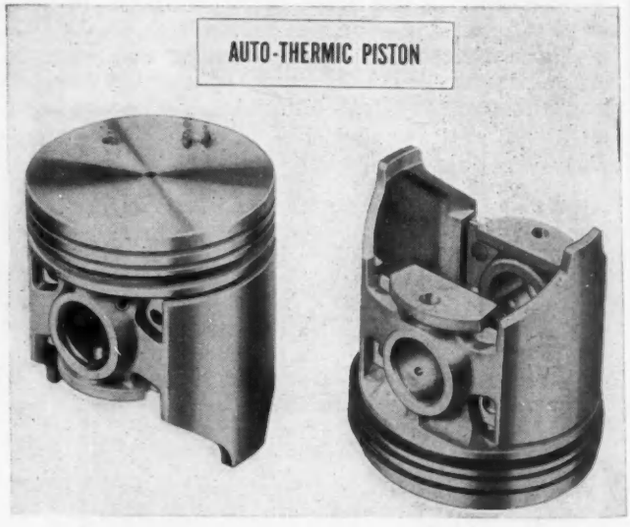
The steel strut, "auto-thermic" design piston greatly reduces piston-to-bore clearance variation due to temperature changes. For example, when the head of the Rocket engine piston is raised from 70 to 400 F, the skirt diameter increases only 0.0038 in. An all-aluminum piston designed for use in this engine increased 0.0074 in. on the same test.

A load deflection comparison between these same two pistons gave 0.0005 in. skirt deflection under 500 lb simulated gas load for the strut piston compared with 0.0035 in. for the all-aluminum design.

The pistons are ground with 0.000 to 0.0005 in. taper and are fit to 0.0005 to 0.001 in. clearance to the bore at the bottom of the skirt. The piston is tin plated to prevent scuffing during break-in.

The two compression rings are both taper-faced and inside beveled. The oil ring is a wide slot, narrow land design to control oil consumption.

Piston weights are equalized in production by removing metal from the pads on the pin bosses. Another innovation in design is the provision for driving and centering the open end of the piston



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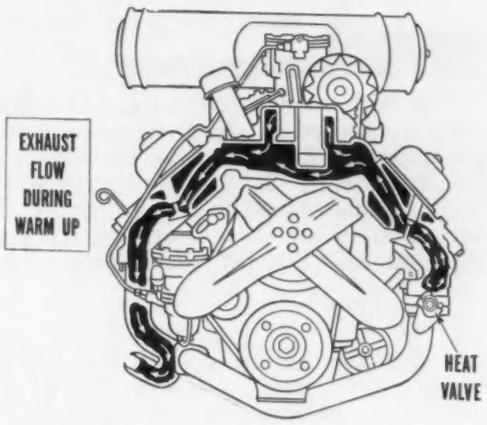
during machining by use of pins in a fixture which fit into holes in the balancing bosses.

Oil economy of the Rocket engine has averaged better than that expected on the 1948 Line-8 engine. In addition to the design of piston and rings described, much work was done on metering and controlling oil to the valve mechanism and preventing its being drawn into the combustion chamber along the valve stems.

Final realization of these results required small rubber seals in a groove on the intake valve stems at the retainers, and control of the volume of oil which ran along the intake rocker arms and down onto the valve stems below the seals.

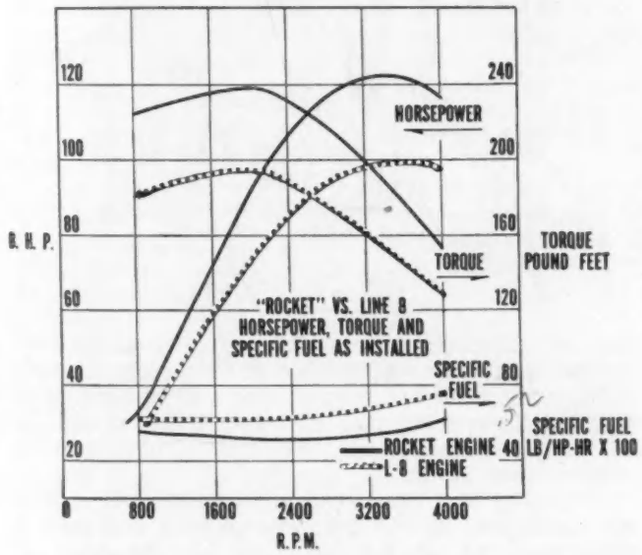


9. Exhaust Flow

The left-hand manifold incorporates a heat valve at the outlet which, when closed during the warm-up period, directs the exhaust gas from that bank through a passage in the center of the head and into the intake manifold heat box. As the diagram at right shows, it is then directed through the heat box where it travels all around the risers and out through the right-hand exhaust manifold.

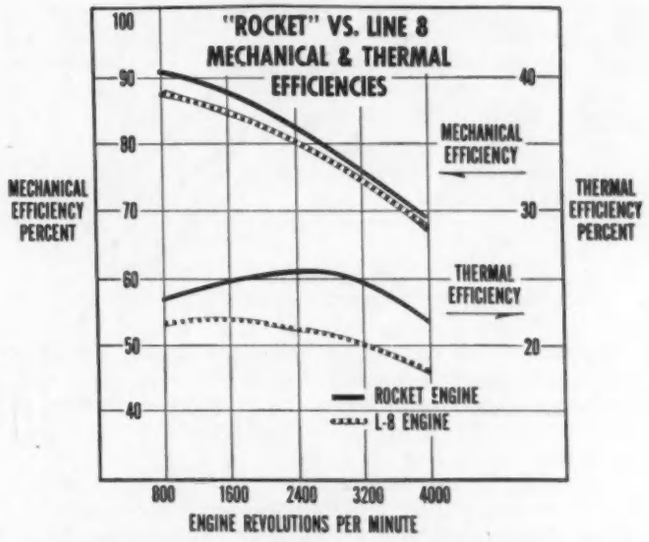
The heat valve incorporates an anti-rattle design feature in which the thermostat spring, the weight, and the stop spring are all placed so that their separate reactions are in the same direction on the one end of the shaft. This load forces one end of the shaft up and the other end down at all times and prevents movement and rattle.

10. Horsepower and Torque



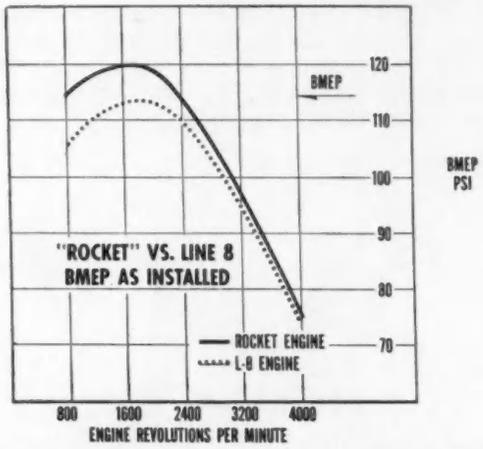
The chart above shows that the "as installed" peak horsepower of the Rocket engine is 122 as compared with 99 of the Line-8. Installed peak torque of the Rocket engine is 240, the Line-8, 194. The Rocket's minimum specific fuel consumption is 0.51 compared with 0.62.

11. Efficiencies



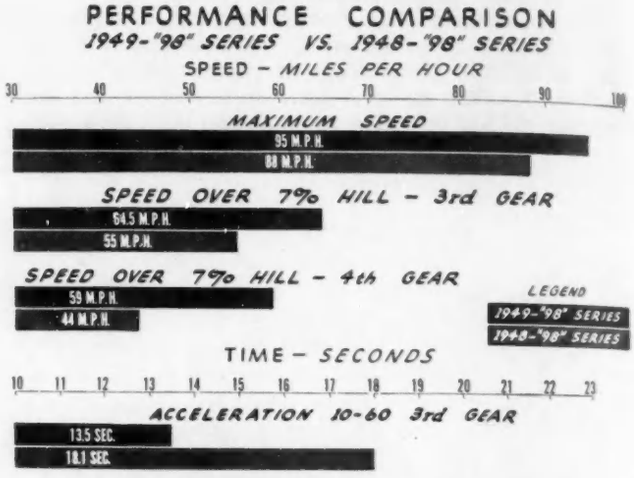
The upper curves in this chart refer to mechanical efficiency and lower ones to thermal efficiency. It can be seen that in both cases substantial gains were made with the new engine compared with the old one over the speed range.

12. BMEP Comparisons



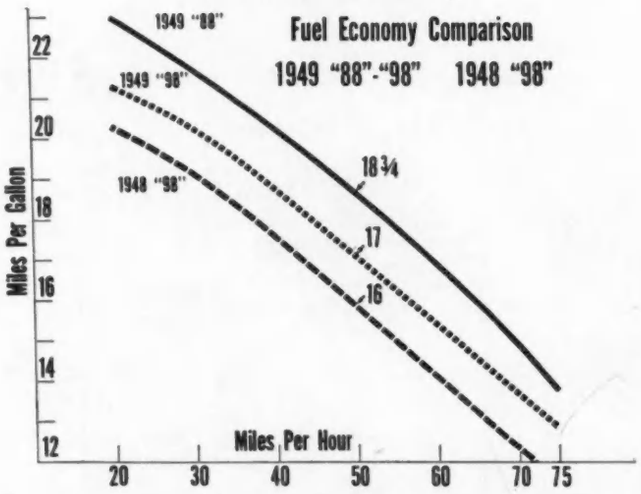
These curves compare the "as installed" bmep of the old and the new engines. Maximum of the Rocket engine is 120, that of the Line-8, 113. High gain at low speeds was deliberately aimed for by valve timing to favor low speed acceleration in fourth gear with Hydra-Matic.

13. Performance



The tests from which these data were obtained were conducted on the Oldsmobile model "98" with the standard 3.63 axle ratio, giving an N/V of 42½. The Oldsmobile "88" has similar performance. But because that model is 300 lb lighter, it has a 3.23 axle ratio, giving an N/V of 38.6. (N represents engine rpm and V is car speed in miles per hour.)

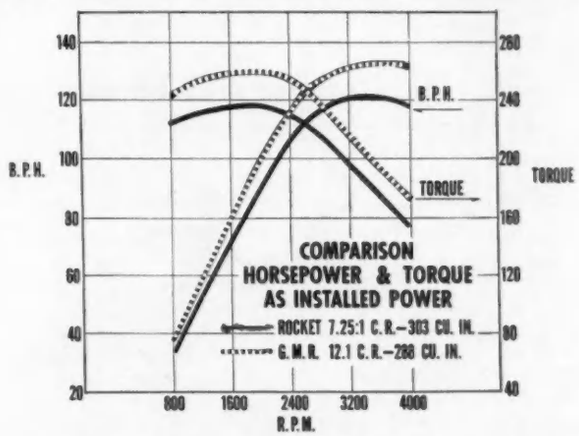
14. Fuel Economy



These curves represent constant speed gasoline economy tests run at the GM Proving Grounds. The lower curve represents the fuel consumption of the 1948 "98," the middle curve the 1949 "98," and the upper curve the 1949 "88."

Increased power and efficiency of the Rocket engine has been used to increase both performance and gasoline economy. The 1949 "98" shows about 1 mpg gain over the speed range. At 50 mph the Line-8 delivers 16 mpg and the Rocket 17 mpg. The "88," with its lighter weight and reduced engine speed due to the low axle ratio, gives an even further gain fuel mileage. It gives 18¾ mpg at 50 mph.

15. At 12 to 1 Compression Ratio



In anticipation of the day when commercial fuels will have been increased in octane rating sufficiently to allow a production engine with a 12 to 1 compression ratio, experimental work has been carried out with revised Rocket engines to determine possible performance gains.

In the 12 to 1 compression ratio Rocket engine, the displacement will be reduced from the present 303 cu in. to 288 cu in. This will be done by shortening the stroke.

As can be seen from the curves in the chart above, horsepower of the engine is increased from 122 to 134 by this increase in compression ratio, despite the smaller displacement. Similarly the torque is increased from 240 to 260 ft-lb.

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NEW MACHINE Evaluates Blasting Material Life

BASED ON REPORT BY

J. F. Ervin

Alloy Metal Abrasive Co.

A new machine that tests blast cleaning shot and abrasives produces an effect comparable to actual blasting operations. The machine, shown in Fig. 1, accurately tests all classes of blasting materials—sum as shot, grit, granet, sand, aluminum oxide, corn cobs, and apricot pits—and determines its breakdown value in blasting practice.

This new machine grew out of considerable agitation in 1945, developed largely through SAE channels, for the standardization of some method for observing the value of materials used for blasting operations. After much discussion, it was generally agreed that some sort of an impact machine, as nearly as possible duplicating operation in actual practice, would be the correct type for proving blasting materials. This was a revival of interest in shot proving that started 12 years ago.

A sectional view of the machine is shown in Fig. 2. It has a 12-in. diameter beater head, corresponding to ordinary blasting machine wheels, driven at a peripheral speed of 200 fps. This is approximately the accepted standard velocity used in cleaning operations.

It has a revolving drum which contains an anvil and lifting device to feed the material through the machine automatically, positively, and constantly at a certain specified rate. The drum, driven by separate power at 25 rpm, feeds the material through at the rate of 25 times per min and removes the broken material to the dust bag.

This machine has an additional motor-driven fan as a dust collector to pick up dust. It is not necessary to use this when testing a material which has a very slow breakdown, because the dust is not serious. However, some sort of suction to discharge dust from the room is available, although not absolutely necessary, with a material which breaks down very rapidly.

Mechanical operation of the machine in proving a material is very simple. First, measure the desired sample by weight or volume, or whatever system is used. Then turn the drum of the machine until the plug is on the top and pour the material into the machine. Close the plug and turn on the power driving the beater head. As soon as it is up to speed, turn power on drum motor which controls feed.

A counter is attached to the machine to measure

ADVENT of a new test machine has simplified and made more accurate the presently laborious job of testing blast cleaning materials for breakdown resistance, says Ervin in this article. It is based on a report he made recently to the Shot Peening Division, of the SAE Iron & Steel Technical Committee.

In describing the machine, he points out that it correctly evaluates metallic and nonmetallic blasting materials because it simulates production blasting. How this test machine works as well as results it produces also are disclosed here by Ervin.

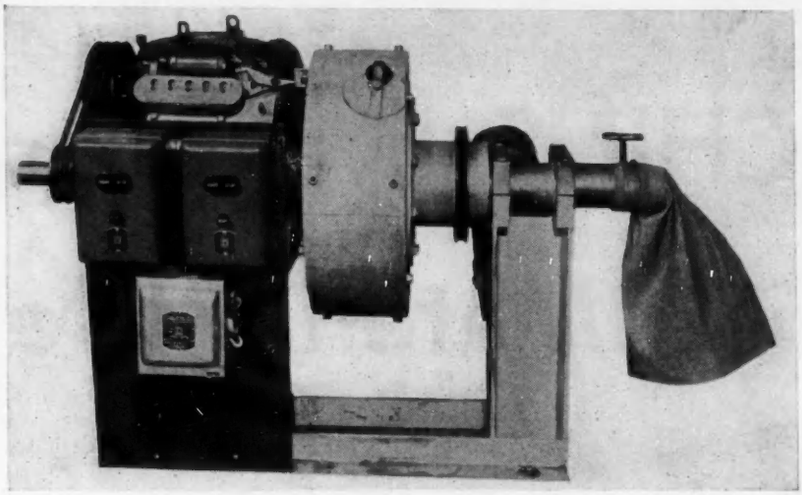

Fig. 1—This new blast cleaning medium machine fulfills practically all the specifications for a machine of this type. For example, it duplicates actual blasting operations, produces accurate data, is practical and rugged, simple to operate, quiet, compact, and dustless

the number of revolutions during this operation. It is advisable to be somewhat familiar with the amount of resistance there is in a type of material to regulate the time for the machine to operate. Operating time which produces 10% loss of material is about the maximum required for each observation to get accurate results. This represents a certain number of passes for each type of material.

On the question of sample size, a standardized sample of 500 g has been adopted largely because screening equipment is used to which this size sample is easily adaptable. Sample size is based largely upon the cubic content, calculated to be about 7 cu in. If different classes of blasting materials are to be measured, the relative specific gravity of these different materials makes it necessary to measure them volumetrically rather than by weight. But for metallic materials, 500 g is a standardized sample.

Now that we know what the machine is and how it works, we must decide on a method of observation. There is no accepted method for charting test results. It has been accepted to some extent that if a sample homogeneous material is placed in a breakdown machine, observations can be made of the sample and the amount of loss of material beyond a

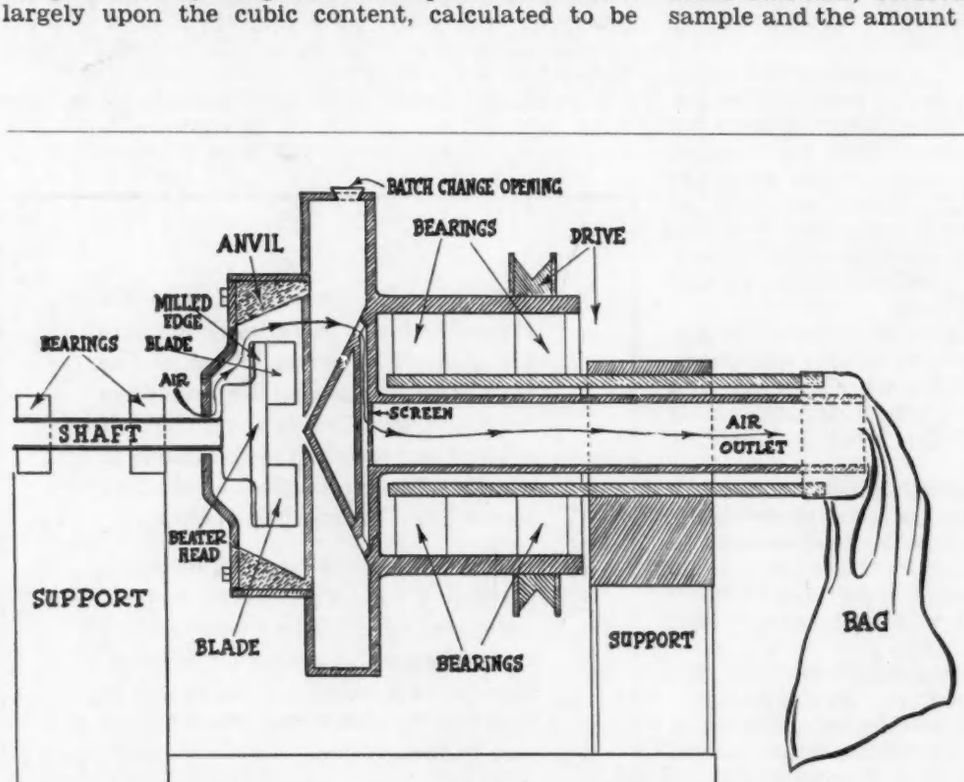

Fig. 2—Sectional view of proving machine

certain screening analysis shall be observed. This is done by taking steps in the breakdown and plotting a chart until 55% of the material has been dissipated below a certain screen analysis.

It is generally accepted that all material is considered as dissipated that will pass through an 0.011 screen opening. Practice in the field has shown that to make a separation between sand and other materials removed from the article being cleaned, all materials below this screen size are normally exhausted into the dust collector or other disposal equipment.

Fig. 3 shows the results of tests made on a number of different types of material selected at random. It points up the wide variation in products offered for blast cleaning and peening operations in resistance-to-breakdown values.

The ordinate shows the weight of material used in the sample, beginning from 200 g up to 500 g, with the base line drawn at 225. This indicates the differential of 275, or 55% loss. The abcissa shows only relative figures. These figures reveal that some materials offered about "1" on this chart of resistance to breakdown, while other materials grade all the way to "plus 18."

Another series of tests throws light on the controversy over value of material size. As shown in Fig. 4, a breakdown of a hard iron material and steel reveals that size of material charged into the machine makes no appreciable difference in life.

The blast cleaning proving machine also can be used to determine the breakdown value of compounded materials. It can be done by testing the compounded materials in proportion of each in the compound and plotting a chart from the resistance of these elements.

When the actual compound make-up is not available, the material must be run through the testing machine for as many passes as will exhaust or break down completely the higher grade product.

Test to Complete Exhaustion

Whether a material is a compound or homogeneous, a specific measuring method must be followed to get its actual resistance to breakdown or to determine how much material must be added to the test machine at all times to keep it in operation. This is it:

First, segregate a sample of a minimum of 1500 g or more. Place 500 g of this sample in a testing machine and run it until no more than 10% of the original 500 g of material has been dissipated. Then, after removing and weighing the material which is left, add enough new material to get the machine up to the original charge, because this duplicates the operation in a commercial blasting machine.

Continue this until 1000 g or more are used by the addition method, measuring the number of passes of breakdown cycles the material resisted during this period. Then use the average added for each 100 cycles for the last 500 g.

Fig. 5 is a chart plotted on the use of 1500 g samples of different types of material to show the average amount of consumption per 100 cycles for the last 500 g added. This shows a correct picture of the actual amount consumed in the machine when the levels are maintained between 90 and 100% of the original charge. Although more time is required for the test, it is an accurate method of measuring the values of the material.

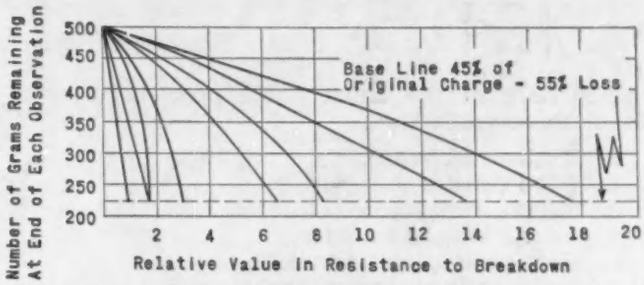

Fig. 3—Chart showing relative breakdown resistance values of grades of metal abrasives. These materials are all SAE Standard sizes. The chart is based on an original sample of 500 g. All material below 0.010 screen size is removed

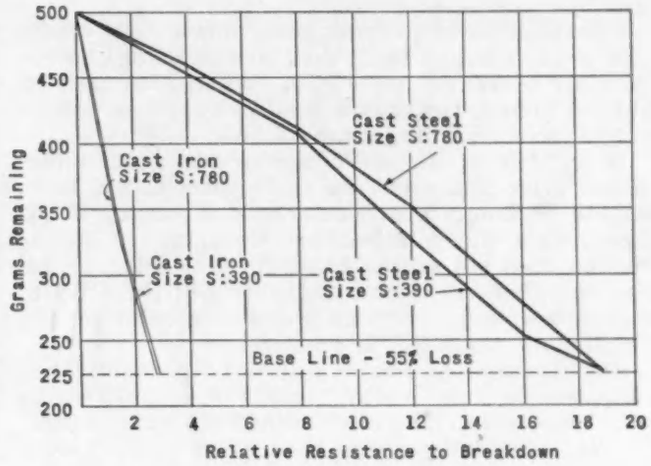

Fig. 4—This chart reveals that size of a material has no appreciable effect on its life

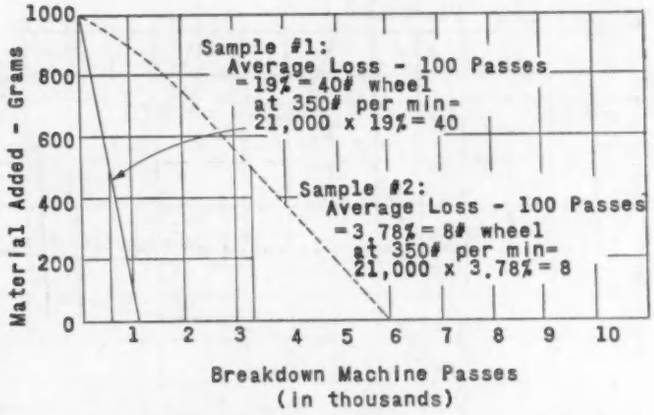
Material Added - Grams
1000
800
600
400
200
0
Sample #1: Average Loss - 100 Passes = 19% = 40# wheel at 350# per min= 21,000 x 19% = 40
Sample #2: Average Loss - 100 Passes = 3.78% = 8# wheel at 350# per min= 21,000 x 3.78% = 8
1 2 3 4 5 6 7 8 9 10
Breakdown Machine Passes (in thousands)

Fig. 5—These are data from a standard breakdown test starting with a 1500 g sample. Material was added each time the original charge was reduced 10%. The chart shows the average reduction in percent of total charge for each 100 passes, based on the last 500 g added

EXCERPTS FROM PAPER* BY

Chester S. Ricker

Detroit Editor,
AMERICAN MACHINIST

Fundamentals of TRUCK

ANALYSIS of the truck brake problem involves these seven factors:

1. Engine resistance,
2. Wind resistance,
3. Rolling resistance,
4. Interia weight transfer,
5. Heat generation and cooling,
6. Brake shoe pressures, and
7. Coefficient of friction.

Engine friction load is one of the most common means of dissipating vehicle kinetic energy when checking speed. It is particularly desirable for long grades where the brakes must be used continuously for a considerable period and, therefore, accumulates heat.

Speeds of 30 to 50 mph, going down 1.5% grades can be maintained fairly constant by using the engine for a brake. See Fig. 1. When it is used on steeper grades, the brakes must be used to a greater extent, first snubbing, then all on.

Many drivers slip down one or two gears under direct drive and "rev" up the engine to get more engine braking. This would be satisfactroy if the driver kept the vehicle speed down so his engine rpm's would not exceed those for which it was designed. However, I understand few of them try to do this and engine service problems assume serious proportions when this is practiced.

Some years ago one engine builder experienced considerable main bearing trouble on engines which were used in trucking over Tehatchapi Pass in California. Excessive engine speeds on down grades caused failure because these trucks were being "let down" the grades with the engine in lower gears. Engines designed to run 2400 rpm maximum under load were turning 4500 rpm when used as a brake.

This was disastrous because centrifugal and inertia forces on reciprocating parts were more than quadrupled. When under power, the higher inertia forces are partially offset by the higher piston pressures and are not so serious; but when running light there is no such offset.

However, engines have been used very successfully as energy dissipators. Before we had modern high efficiency power brakes, the Swiss-built Sauer trucks were designed to use the engine for a brake in mountain service. To do this, a camshaft was provided with supplementary cams and adjustable axially to turn the engine into an air compressor when it was being used as a retarder.

Wind and rolling resistance help little in retarding heavy trucks at safe down-grade speeds. About 30 hp maximum is what might be expected at 40 mph. This is less than 10% of the force required to hold back a truck at 40 mph on a 6% grade.

In passenger car design, it is possible to estimate the fourth factor in brake analysis, inertia weight transfer to the front wheels when stopping a car at high speed. For this reason car front brakes usually

* Paper "Trends in Heavy Duty Brakes," was presented at SAE National Transportation Meeting, Cleveland, March 30, 1949.

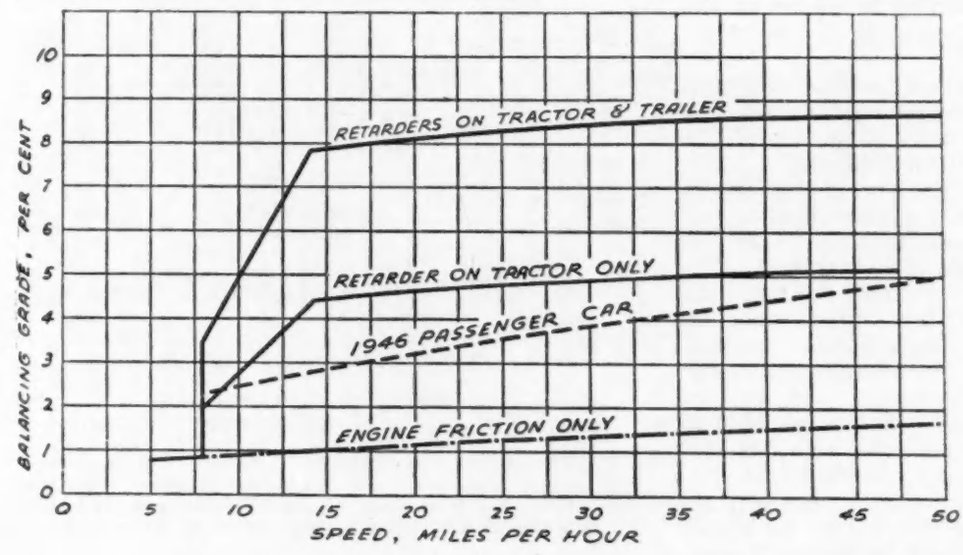

Fig. 1—Balancing grades, which show what a small part of the retardation is due to engine friction and wind resistance

# of BRAKING

are made more powerful than the rear. But on trucks, the arrangement of the load, its position and relationship with varying wheelbases is indeterminate. Therefore, inertia weight transfer cannot be foretold accurately, except for specific installations; and it is less important because of lower speeds.

In connection with weight distribution, it is interesting to note that front axle loading has been increasing by moving the front axle further aft on several new trucks. This allows front brakes to bear a greater proportion of the load. At present, the design weight distribution on the front axles of one line of heavy trucks varies from 23 to 30%. This is detailed in Table 1. It may go even further if power steering were introduced.

How much heat is dissipated in stopping the vehicle is another phase of the problem. If the stops are not too close together, the brakes will cool off enough to stay below a critical temperature. These same brakes, however, might be inadequate on a long grade if the truck operator goes down at 30 to 40 mph.

Just to show the extent of the truck heat problem, a heavy truck column has been added to a table developed several years ago showing what happened when a car, bus, and B-29 airplane were brought to a stop. See Table 2. It clearly shows why bus brakes are usually adequate—the energy that has to be handled per pound of drum weight is about half that of the heavy truck.

The data given are for an old type bus without aluminum body, but would be better today. On the other hand, while the truck has to dissipate more heat, it has much greater braking area if both truck and trailer brakes take their share of the load.

Another picture of brake drum temperatures is given in Fig. 2, which shows the effect of drum loading.

Energy dissipation is the heart of the brake problem. No matter how the energy is absorbed, eventually it is dissipated as heat. Judging from the passenger car brake temperature curves in the paper by Chase[1], it takes from eight to ten times as long to dissipate the heat generated in stopping from 60 mph as to introduce it into the brake drums. Therefore, the problem is to have a cooling cycle between brake applications which will restore brake drums more rapidly to the original operating temperature.

In most installations, slip stream air circulation is depended upon to restore the drum temperature to normal. Brakes have been constantly increased in size up to the limits permitted by the wheel rims. Twenty-inch wheels with wide-drop center rims limit brake diameters to 16 in. These drums are completely buried inside the wheel rim with a narrow annular ring of dead air around them. They cannot cool off quickly enough. In fact, they stay hot so long that they heat up the wheel rims.

If brake drums could be cooled a little better, perhaps the greater proportion of users would never have any trouble. American "Tru-Stop" and Budd railway disc brakes are the ideal design for good air

[1] See SAE Quarterly Transactions, Vol. 3, January 1949, pp. 26-40: "Passenger-Car Brake Performance—Limitations and Future Requirements," by T. P. Chase.

Table 1—Weight Distribution and Brake Capacity

Rated Capacity		1½ Ton	2 Ton	1½ Ton	2 Ton	2½ Ton	3 Ton
G.V.W., lb		14,500	16,500	14,750	16,750	18,500	21,000
Front Axle Capacity		3750	4500	4500	4500	6000	6000
Front Axle % of total		23.0	25.7	26.4	25.7	30.0	27.3
Rear Axle Capacity		12,500	13,000	12,500	13,000	14,000	16,000
Rear Axle % of total		77.0	74.3	73.6	74.3	70.0	72.7
Sum of Capacities		16,250	17,500	17,000	17,500	20,000	22,000
Weight per wheel	front	1875	2250	2250	2250	3000	3000
	rear	6250	6500	6250	6500	7000	8000
Tire Size	front	6.50/20	7.50/20	6.50/20	7.50/20	7.50/20	8.25/20
	rear	7.50/20	8.25/20	7.50/20	8.25/20	8.25/20	9.00/20
Brake Drum	front	14⅛ × 2	16 × 2.5	14⅛ × 2	16 × 2.5	16 × 2.5	16 × 2.5
(diameter × width)	rear	16 × 3	16 × 3	16 × 3	16 × 3	16.25 × 3.5	16.5 × 4
Total Torque in.-lb	front	37,200	43,900	37,200	43,900	43,900	52,700
(Maximum)	rear	52,600	65,500	52,600	65,500	.....	.....
	total	89,800	109,400	89,800	109,400	.....	.....
Total Torque Tractive Effort Ratio (Dry pavement) (f = 0.7)	front	0.88	0.80	0.73	0.80	0.59	0.69
	rear	0.34	0.40	0.34	0.40		
Total Torque Tractive Effort Ratio (wet pavement) (f = 0.4)	front	1.54	1.40	1.28	1.40	0.69	0.81
	rear	0.60	0.70	0.60	0.70		

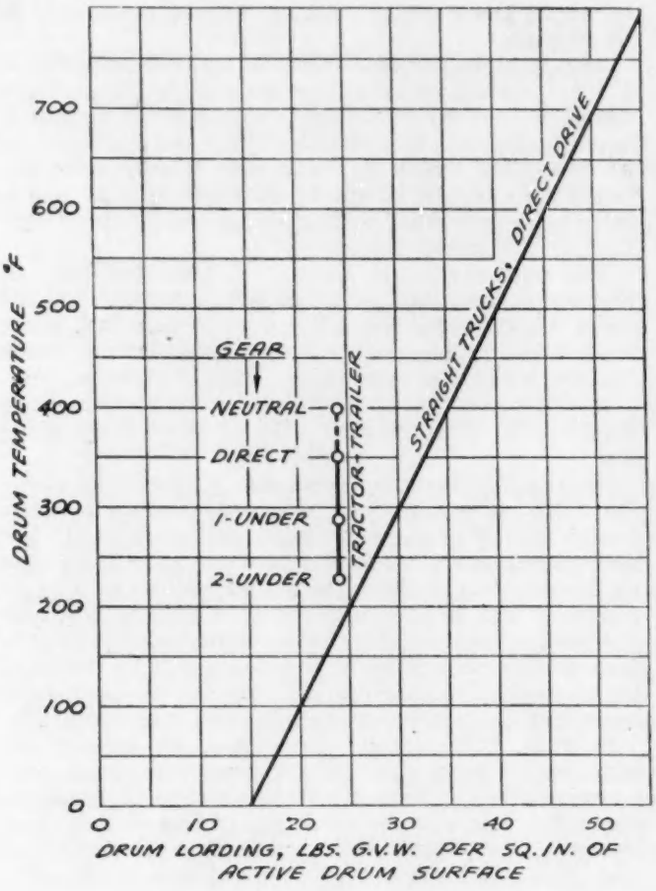

Fig. 2—How drum loading affects drum temperature

cooling. They do not lend themselves to installation on wheels; but I'm inclined to believe that if centrifugal blowers could be cast in the outer edge of brake drums, a lot could be done toward keeping brake drums cooler. This might require moving the rear springs inward an inch or so and the wheels outward.

With power brake applications, the question of brake shoe pressures—sixth brake problem factor—is a matter of engineering choice, limited by the torque desired. For longest life, unit pressures should be between 100 and 250 psi. If the pressure is too low, the brake lining may glaze and if too high, short-lived.

Hydraulic line pressures of 1500 psi are currently good practice, giving actuating pressures from 1500 to 2600 lb or more. Cam-operated brake shoes with air pressure actuation have equally high shoe pressures. Note that this only considers the shoe actuating pressure, not the effective pressure obtained when the wrapping effect is added. This will vary with each installation.

Next step in the analysis of any brake design involves both brake shoe friction and tire tractive friction. It has been the practice to assume 0.7 as the average coefficient of friction between the tire and a cement road, and 0.3 as a good average for brake lining. T. P. Chase shows, in the paper previously referred to, that while 0.7 may be satisfactory on dry pavements, 0.4 is a high average on wet roads.

With this in mind, I extended Table 1 to include maximum torque available on front and rear brakes, using 0.3 as the coefficient of friction. Then using the wheel load and rolling radius of the respective wheels, computed the tractive resistance that could be obtained when using both 0.7 and 0.4 as the tire coefficient of friction.

The ratio between brake torque that could be applied to the wheel and the tire tractive resistance that could be obtained on both dry and wet pavements is shown at the bottom of Table 1. From these it is obvious that on dry pavement, brakes cannot lock the wheels of a loaded truck even in an emergency stop. But wet slippery pavements are another reason why certain western truckers are disconnecting front wheel brakes.

Table 2—Heat and Energy Comparisons

Type of Unit	Passenger Car*	Truck & Trailer	Bus*	B-29 Airplane*
Load Weight, lb	4190	45,750	27,150	140,000
Brake Size & No.	11 × 2(4)	F 16.25 × 4.5(2) R 16.25 × 7.0(2) T 16.50 × 7.0(2)	F 14.5 × 6(2) R 14.5 × 4(4)	20 × 2.75(16)
Swept area, sq in.	277	1902	1277	2760
Speed, mph	70	40	70	110
KE (ft-lbs) total × 1000	685	2440	4430	57,000
KE per sq in. area	2470	1280	3460	20,600
Weight of brake drums, lb	50 Estimated	220 Estimated	640 Estimated	485 Actual
KE per lb of drum weight, ft-lb	13,700	11,100	6925	117,500
Stopping Distance, ft	527	172	527	1303
Total Btu developed	877	3125	5700	73,400
Time of stop, sec	10.28	5.87	10.28	16.20
Btu liberated per sec	85.2	532	555	4530
Btu/sec/sq in swept	0.308	0.280	0.435	1.64

* Information obtained from H. H. Kerr, U.S. Rubber Co., March 5, 1947.

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COMMON ALLOY STEELS

. . . . Their Characteristics & Uses

This article tells engineers about metallurgical properties of commonly-used alloy steels (excluding stainless and austenitic) as well as some of their applications. It was prepared by the Alloy Steels Division, of the SAE Iron & Steel Technical Committee, for publication in the 1950 SAE Handbook.

BY common custom steel is considered as an alloy steel when the maximum of the range given for the content of alloying elements exceeds one or more of the following limits:

- Manganese 1.65%
- Silicon 0.60%
- Copper 0.60%

or, steel for which definite range of chromium up to 3.99%, nickel, molybdenum, vanadium, or a specified minimum quantity of any of the following, or other useful elements, is implied or required: aluminum, boron, cobalt, columbium, titanium, tungsten, or zirconium.

The principal uses for alloying elements in the common constructional steels are: (1) to overcome the effect of mass in hardening, permitting use of a slower quench than that obtained with water and thus avoiding excessive distortion and cracking, and (2) to a limited extent, for obtaining some special qualities not possessed by a simple carbon steel of equivalent carbon content.

Alloy steels are generally not specified for use without appropriate heat-treatment. Some special properties which certain alloying elements impart are: retardation of transformation, lowering of transformation temperature, resistance to creep at elevated temperatures, retention of toughness at sub-zero temperature, resistance to wear, and effect on hardness and machinability.

Other special properties may be imparted by the proper choice of alloying elements. Thus, when the proper percentage and type of alloying elements are chosen, a complicated part, after being completely austenitized, may be hardened by quenching at a temperature considerably below the austenitizing temperature, and thereby greatly reduce the usual distortion and residual stresses accompanying the quenching.

Alloy steels are usually made under the best practice which will yield fully killed steel in the ingot. Such practice generally assures uniformity of composition and hardenability.

The hardenability or response to heat-treatment is probably the most important criterion for the selection of steel. By the term "hardenability" is meant the property that determines the depth and distribution of hardness induced by quenching from above the transformation range. It is well known and easily proved that the surface hardness of all steels, including the common alloy steels, when quenched at the optimum speed for the particular composition depends upon the carbon content *at the surface.*

The optimum speed for the production of maximum hardness, however, may be greater than that produced by oil quenching. Consequently hardenability as well as carbon may affect the surface hardness of medium carbon, oil-quenched alloy steels.

Decarburized, scale-coated, and overheated surfaces, or surfaces from which heat is abstracted at less than the critical speed, may not attain expected hardness. When a carbon content of 0.65% is reached, little more may be expected in terms of indentation hardness (Rockwell or Brinell), regardless of the quenching practice or the alloy content of the steel. Most of the physical properties, such as tensile strength, hardness, and toughness are related to the thoroughness with which the transformation in hardening is effected.

By "hardening" it is implied that a different structure has been obtained than that originally existing in the piece. The hardest type of structure is martensite. The hardness of the 100% martensitic structure depends wholly upon the carbon content. If the heat-treatment is such that a minimum of 90% martensite is produced in quenching, and such that the final hardness is produced with proper tempering, it may be expected that all the final physical properties—commonly computed as tensile strength, yield strength, and fatigue strength—will be dependent upon the surface and cross-sectional hardness.

If, in commercial operations, a high percentage of martensite is not obtained in quenching, a cor-

responding reduction in the final properties may be expected after the tempering operation.

During the present decade, it has been fairly well established that the physical properties obtainable in a given section of steel are very closely evaluated by the measure of hardenability attained in the particular section, irrespective of the steel composition. That is to say, many different steel compositions can be interchanged to obtain a similar range of physical properties in the finished part if these physical properties are properly ascertained by careful consideration of the hardenability test.

The SAE Standard End-Quenched Hardenability Test, commonly called the "Jominy," has enabled the investigator to compare easily and certainly, the characteristic hardenability of various types of steel composition or steels of the same typical composition made by different practices. In other words, the end-quenched hardenability test is a convenient summation of all the characteristics which determine the degree of a surface hardness and the depth of penetration of hardness in quenching. By tempering the end quenched bar, the final hardness of the product, after quenching and tempering, may be predicted.

Properties such as ease of annealing, degree of hardening from cold working, ease of machining, and some other characteristics may be peculiar to either composition or heat-treatment, or both. The choice of steel for a given application is more often dictated by the overall economic consideration than by any other factor.

In general, the common automotive alloy steels may be divided into two grades: (1) the low-carbon carburizing and (2) the higher carbon directly hardenable grade.

Carburizing Grades of Alloy Steel

The properties of carburized and hardened cases depend upon the carbon and alloy content, the structure of the case, and the degree and distribution of residual stresses. The carbon content of the case depends upon the details of the carburizing process, and the response of iron and the alloying elements present to carburization. The original carbon content of the steel has little or no effect upon the carbon content produced in the case.

The hardenability of the case therefore, depends upon the alloy content of the steel and the *final* carbon content produced by carburizing, but not upon the *initial* carbon content of the steel.

With complete carbide solution, the effect of alloying elements upon the hardenability of the case will in general be the same as the effect of these elements upon the hardenability of the core. As an exception to this, any element which inhibits carburizing may reduce the hardenability of the case. It is also true some elements which raise the hardenability of the core may tend to produce more retained austenite and consequently somewhat lower hardness in the case.

Alloy steels are frequently used for case hardening because the required surface hardness can be obtained by moderate speeds of quenching. This may mean less distortion than would be encountered with water quenching. It is usually desirable to select a steel which will attain a minimum surface hardness of 58 or 60 Rockwell C after carburizing and oil quenching. Where section sizes are large, a high hardenability alloy steel may be necessary, while for medium and light sections, low hardenability steels will suffice.

In general, the case hardening alloy steels may be divided into two classes so far as the hardenability of the case is concerned. Only the general type of steel, 3300–4100, and so forth, is given. As the original carbon content of the steel has no effect upon the carbon content of the case, the last two digits in the specification numbers are not meaningful so far as the case is concerned.

As SAE, 2500, 3300, 4300, 4800, and 9300 are high alloy steels, both the case and the core have high hardenability. These types of steel are used particularly for carburized parts having thick sections, such as bevel drive pinions and heavy gears. Good case properties can be obtained by oil quenching. These steels are likely to have retained austenite in the case after carburizing and quenching, consequently special precautions or treatments, such as refrigeration, may be required.

Carburized cases of SAE 1300, 2300, 4000, 4100,

Table 1—Breakdown of Case-Hardening Steels by Core Hardenability

1. Low Hardenability Core	2. Medium Hardenability Core	3. High Hardenability Core
4017	1320	2517
4023	2317	3310
4024	2512	3316
4027*	2515*	4320
4028*	3115	4817
4608	3120	4820
4615	4032	9310
4617*	4119	9315
8615*	4317	9317
8617*	4620	
	4621	
	4812	
	4815*	
	5115	
	5120	
	8620	
	8622	
	8720	
	9420	

* Borderline classifications; might be considered in next higher hardenability group.

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4600, 5100, 8600, and 8700 steels have medium hardenability. This means that their hardenability is intermediate between that of plain carbon steel and the higher alloy carburizing steels just described. In general, these steels can be used for average size case-hardened automotive parts, such as gears, pinions, piston pins, ball studs, universal crosses, crankshafts, and so forth. Satisfactory case hardness should be produced in most cases by oil quenching.

Core Properties

The core properties of case-hardened steels depend upon both carbon and alloy content of the steel. Each of the general types of alloy case-hardening steel is usually made with two or more carbon contents so as to produce different hardenability in the core.

The most desirable hardness for the core depends upon the design and functioning of the individual part. In general, where high compressive loads are encountered, relatively high core hardness is benefical in supporting the case. Low core hardnesses may be desirable where great toughness is essential.

The case-hardening steels may be divided into three general classes depending upon hardenability of the core. See Table 1. For hardenability of individual steels, see hardenability specifications for "H" steels. Due to the fact that "H" bands have not been worked out for all steels, it is impossible to give an accurate comparative rating of hardenability of all the steels in any one group.

In general, all of the alloy carburizing steels are made fine grain and most are suitable for direct quenching from the carburizing temperature. Several other types of heat-treatment involving single and double quenching are also used for most of these steels.

Directly Hardenable Grades of Alloy Steels

These steels may be considered in five groups on the basis of approximate mean carbon content of the SAE specification. In general the last two figures of the specification agree with the mean carbon content. Consequently the heading "0.30–0.37 Mean Carbon Content of SAE Specification" includes steels such as 1330, 3135, and 4137.

Mean Carbon Content of SAE Specification	Common Applications
(1) 0.30 – 0.37	Heat-treated parts requiring moderate strength and great toughness.
(2) 0.40 – 0.32	Heat-treated parts requiring higher strength and good toughness.
(3) 0.45 – 0.50	Heat-treated parts requiring fairly high hardness and strength with moderate toughness.
(4) 0.50 – 0.62	Springs and hand tools.
(5) 1.02	Ball and roller bearings.

It is necessary to deviate from the above plan in the classification of the carbon molybdenum steels. When carbon-molybdenum steels are used, it is customary to specify higher carbon content for any given application than would be specified for other alloy steels, due to the low alloy content of these steels.

For example, 4063 is used for the same applications as 4140, 4145, and 5150. Consequently in the following tables and discussion, the carbon molybdenum steels have been shown in the groups where they belong on the basis of applications rather than carbon content.

For the present discussion, steels of each carbon content are divided into two or three groups on the basis of hardenability. Transformation ranges and consequently heat-treating practices vary somewhat with different alloying elements, even though the harenability is not changed.

Steels with a 0.30 – 0.37 mean carbon content of SAE specification are frequently used for water-quenched parts of moderate section size and for oil quenched parts of small section size.

Typical applications of these steels are connecting rods, steering arms and steering knuckles, axle shafts, bolts, studs, screws, and other parts requiring strength and toughness where section size is small enough to permit obtaining the desired physical properties with the customary heat-treatment.

Steels falling in this classification may be subdivided into two groups on the basis of hardenability:

Hardenability Classification—0.30 – 0.37 Carbon

(a) Low Hardenability		(b) Medium Hardenability	
1330	4042	2330	
1335	4130	3130	
	5130	3135	8635
	5132	4137	8637
4037	8630	5135	8735
		8632	9437

In general, steels with a 0.40 – 0.42 mean carbon content of SAE specification are used for medium and large size parts requiring high degree of strength and toughness. The choice of the proper steel depends upon the section size and the mechanical properties which must be produced. The low and medium hardenability steels are used for average size automotive parts, such as steering knuckles, axle shafts, propeller shafts, and so forth. The high hardenability steels are used particularly for large axles and shafts and for large aircraft parts.

These steels are usually considered as oil quenching steels, although some large parts made of the low and medium hardenability classifications may be quenched in water under properly controlled conditions.

These steels may be roughly divided into three groups on the basis of hardenability:

Hardenability Classification—0.40 – 0.42 Carbon

(a) Low Hardenability		(b) Medium Hardenability		(c) High Hardenability
1340		2340	4640	4340
4047	5140	3140	8640	
	9440	3141	8641	9840
		4053		
		4063	8642	
		4140	8740	
			8742	
			9442	

Continued on page 60

VARIETY of Compound Engine

BASED ON PAPERS* BY

E. J. Manganiello, L. V. Humble, and D. S. Boman

NACA LEWIS FLIGHT PROPULSION LABORATORY

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(These papers will be printed in full in SAE Quarterly Transactions.)

"COMPOUND POWERPLANT" is a term that covers many configurations of many types of three components: supercharger, reciprocating engine, and means for taking energy from the engine component's exhaust. Major differences between various configurations lie in the disposition of engine and turbine power between supercharging and useful power.

Basically, the compound powerplant consists of a supercharged reciprocating engine with means for utilizing exhaust power.

Usually the supercharger is an axial or centrifugal compressor, but it may reciprocate with the engine. The engine may be either spark ignited or compression ignited, four-stroke or two-stroke. The means for using exhaust gas energy is most often the gas turbine—either a steady-flow turbine or a blowdown (puff-energy) turbine, or both. Alternately, jet stacks have been used successfully on aircraft and secondary low-pressure cylinders less successfully on stationary engines.

Steady-flow turbines develop power from the pressure energy of the exhaust gas. They impose a higher-than-atmospheric back pressure on the reciprocating engine component. Blowdown turbines derive power from the velocity energy generated during the exhaust blowdown period. They require individual stacks from each cylinder or groups of cylinders without overlapping exhaust periods to the turbine nozzle box. Because they utilize the existing velocity energy of the exhaust gases, they do not impose any back pressure on the engine. Where both types of turbine are used on one compound powerplant, exhaust gases are ducted to the blowdown turbine first.

The high cycle temperatures attained in the reciprocating engine coupled with the fuller expansion permitted by the turbine give the compound engine better thermal efficiency than is had by the reciprocating engine alone or by the conventional gas turbine powerplant. The reciprocating engine can tolerate high cycle temperatures because they prevail for a very short time only. Coolant surrounding the cylinder walls cools the metal, and expansion cools the gases to temperatures the turbine can withstand. The turbine is a lightweight, compact means of taking out exhaust energy from large volumes of gas.

For a given combination of supercharger, reciprocating engine, and turbine, the power of the various components varies with engine exhaust pressure. With increasing ratio of reciprocating engine exhaust pressure to manifold pressure, supercharger power requirement decreases due to decreasing volumetric efficiency of the engine, engine power output decreases as a result of decreasing volumetric efficiency and increasing pumping losses, steady-flow turbine power output increases primarily because of the increasing expansion ratio and secondarily because of increased turbine inlet

* Manganiello-Humble-Boman paper, "Compound Engine Systems for Aircraft," was presented at the SAE Summer Meeting, French Lick, Ind., June 6, 1949. Schweitzer-Salisbury paper, "Compound Power Plants," was presented at the same meeting, on June 7. (These papers are available in full in multilithographed form from SAE Special Publications Department. Price 25¢ each to members, 50¢ each to nonmembers.)

Schemes Possible

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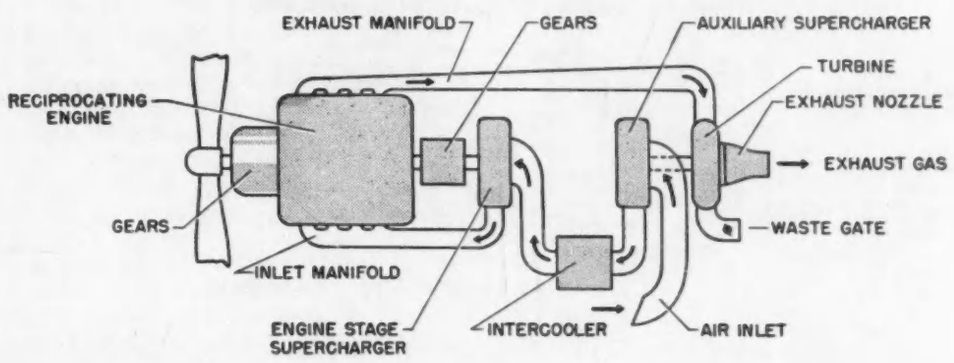
GENERAL ELECTRIC CO.

temperature (the decrease in volumetric efficiency reduces the increase), and blowdown turbine power output decreases because of decrease in available kinetic energy and volumetric efficiency.

Aside from differences in type of supercharger, reciprocating engine component, and turbine, there are differences in arrangement of the three components. In some configurations, the reciprocating engine delivers all the useful power, and the turbine merely powers the supercharger. In others, both reciprocating engine and turbine contribute to the shaft power. And in still others, the reciprocating engine powers the compressor only, and the turbine supplies all the shaft power.

Examples of these three types of configurations follow. Sketches are schematic only.

1. Engine Delivers All Useful Power



1A. Turbosupercharged Aircraft Engine—In the conventionally turbosupercharged aircraft engine, a steady-flow turbine provides only enough power to drive an auxiliary or first-stage centrifugal supercharger. Engine exhaust gas is ducted to the turbine, which drives the auxiliary supercharger. The engine charge air is compressed by the auxiliary supercharger, cooled by the intercooler, and then further compressed by the engine-stage supercharger to intake manifold pressure. From the manifold, it goes to the engine cylinder.

Because the exhaust gas may contain more avail-

able energy than is required to drive the auxiliary supercharger, the turbine is provided with a waste gate for bypassing part of the exhaust gas.

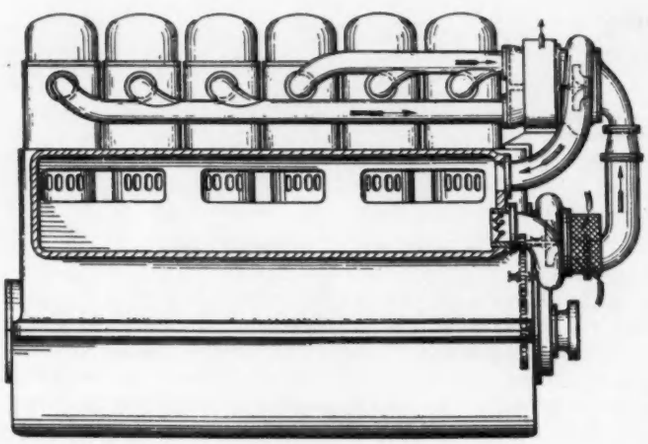
Rearwardly directed jet nozzles recover energy from the turbine leaving velocity.

Substitution of an aftercooler for the intercooler in this configuration would result in lower inlet manifold temperature. This would increase breathing capacity and raise knock limits on manifold pressure. Both these factors would contribute to higher output. But practical installation problems interfere with application of aftercoolers to existing radial engine systems with built-in engine-stage superchargers. There is no room for the aftercooler between the engine stage supercharger and the manifold.

In an aircraft engine designed from scratch for compounding, it might be possible to power both first- and second-stage superchargers by turbine and provide for aftercooling.

Rateau and Moss introduced turbosupercharging to aircraft engines during World War I. Its principal advantage is that it compensates for power loss at altitude due to decreased air density.

1B. Buchi Duplex-Turbocharged Diesel Engine—Buchi has proposed a two-stroke cycle diesel engine compounded with a geared blower and a turbocharger. The two centrifugal-type superchargers would work in parallel at low speeds, but change automatically to series operation when the turbocharger pressure exceeds the geared-blower pressure.

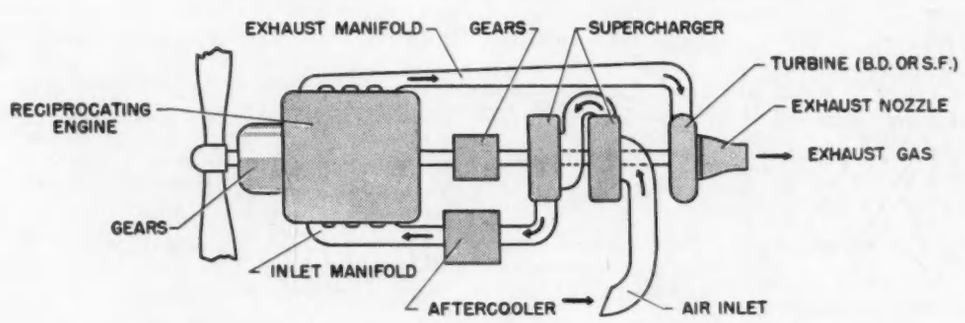


Because the two-stroke cycle diesel is not self-breathing, it must be supercharged at all speeds. At low speeds, turbine power output and consequently turbine-driven supercharger air output is slight. So the job is done by a gear-driven blower that can deliver air at the low as well as the high end of the speed range. (The gear-driven supercharger is not needed on turbocharged four-stroke diesels because they are self-breathing and need supercharging only at high load, where adequate turbine power is available.)

The turbine is geared to its blower only; there is no power transmission from turbine to engine shaft.

Only compression-ignition engines can take full advantage of supercharging. The Duplex-Turbocharged diesel would enjoy the advantages of the two-stroke cycle as well.

2. Engine *and* Turbine Deliver Useful Power



2A. Geared-Turbine Supercharged Powerplant[1]—Introduction of a shaft and gearbox between the turbine and the reciprocating engine enables the turbine to contribute power in excess of supercharger demands to the shaft output (shown with two-stage supercharging).

American engineers have found configurations of this general type suitable for aircraft powerplants. The Swiss use it in a marine diesel. They connect a multistage axial compressor-turbine set to the crankshaft through gearing and an hydraulic coupling.

[1] See SAE Journal, Vol. 57, No. 9, September, 1949, pp. 39-42: "Compounding the Piston Engine," by F. J. Wiegand and M. R. Rowe.

2B. Parallel-Engine Arrangement—A reciprocating engine may be added to a turboprop between compressor and combustor so that, while some of the gas passes through the two components in series, power delivery is parallel. The reciprocating engine is supercharged by the turboprop compressor and exhausts into a venturi at the entrance to the turboprop combustor.

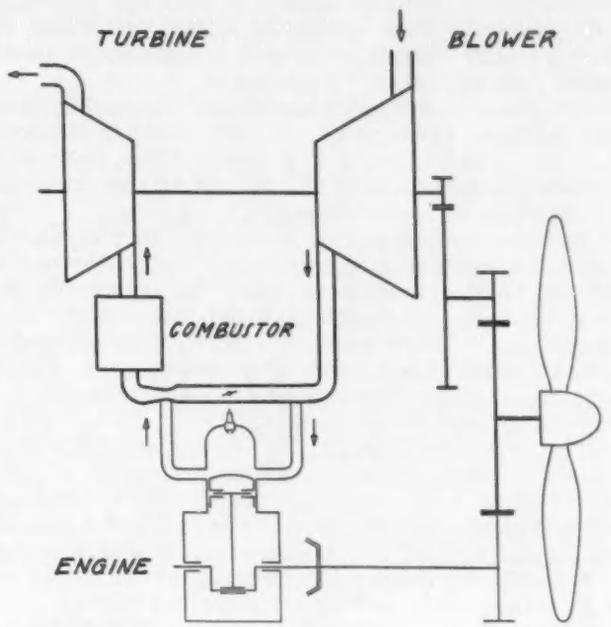
Allowable combustor inlet temperature may limit the proportion of engine exhaust gas to compressor air. If necessary, engine exhaust can be introduced directly at the turbine nozzles.

Part of the fuel supplied is burned in the reciprocating engine, producing power at higher efficiency than in the simple turboprop. Part is burned in the turboprop combustor with 300-400% excess air to keep final gas temperature down to levels the turbine can stand. Unburned fuel and carbon monoxide escaping from the engine burn in the combustor.

The proportion of power generated by the reciprocating engine may be large or small, depending upon the design goal. If compactness is the goal, the engine may be no larger than required for starting the turboprop. If emphasis is on fuel economy, the engine may develop a substantial proportion of total power. The ratio can be changed in flight by moving a butterfly valve in the air duct supplying the reciprocating component.

Considerable work was done by a German concern during the war on an aircooled, valveless, two-stroke-cycle engine.

Advantages of this type of reciprocating component are its large air-handling capacity, simplicity, compactness, and avoidance of valves. Because of its piston-controlled ports, the engine is less prone to detonation and the faster blowdown yields greater recovery.



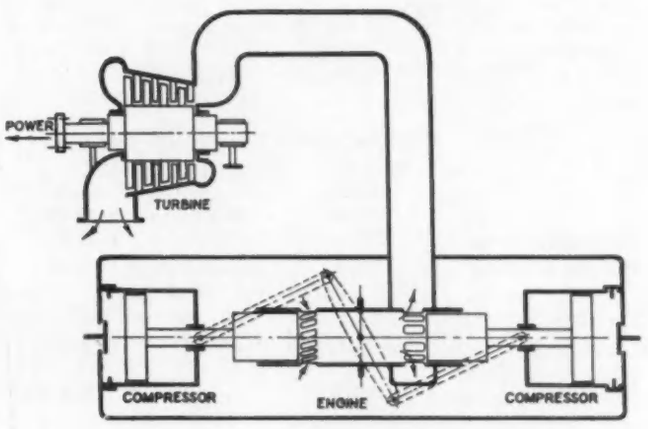
3. Turbine Delivers All Useful Power

3A. Free-Piston Gas Generator Plus Turbine[2]—Compressor and reciprocating engine may be combined into one power-balanced gas-generating unit which serves only as a source of high-temperature, high-pressure gas for a turbine.

In this arrangement, there is no exchange of power between the engine and the turbine. All the engine power goes into supercharging. All the turbine power is available for doing useful work.

Engine power output balances economically with compressor power demand at supercharge pressure ratios around 6 or 7. These ratios are high enough to preclude their use in today's spark-ignition engines, but they are advantageous in compression-ignition engines.

French, British, and American engine builders use opposed free pistons in their diesel gas generators. Compressor pistons are integral with power pistons. When combustion pressures push the two piston assemblies apart, the power pistons uncover exhaust ports from which the gases pass to the turbine, and the compressor pistons compress air for scavenging



the cylinder and for the next charge. Scavenge air also goes to the turbine.

In true free-piston designs, there is no crankshaft. Energy for the return stroke is stored in air compressed by a third piston acting in a "bounce" chamber. This bounce energy returns the pistons to the position where compression ignition again takes place.

[2] See SAE Journal, Vol. 57, No. 10, October, 1949, pp. 62-64: "Free-Piston Gasifier Conserves Turbine Alloys," by L. F. Small.

(This paper will be printed in full in SAE Quarterly Transactions)

Synthetics Help

PERFORMANCE limitations of petroleum lubricants at extremely high and low temperatures lead to investigation of synthetic lubricants for the aircraft of our modern global air force. The result is that the improvements in the specifications for a grease and a light oil resulting from the properties of synthetics gave materials which performed reliably under conditions in which comparable petroleum products proved inadequate.

Air Force and Navy aircraft are required to operate reliably throughout the temperature range of - 67 to + 160 F. For many applications, petroleum lubricants satisfactory at one end of the range are inadequate at the other end.

So the military services working in cooperation with the petroleum and chemical industries turned to synthetic lubricants—that is, nonpetroleum materials, or compositions which are largely nonpetroleum.. They studied esters of dibasic acids, polyalkylene glycols and their derivatives, silicon organics, halogenated hydrocarbons, and inorganic-radical organic-base esters.

The esters of dibasic acids have brought the most success so far. They form the basis for two Air Force-Navy Aeronautic Specifications: AN-G-25, a low-temperature low-volatility grease and AN-O-11, an instrument oil. Although straight esters proved unsuitable for engine oils in tests, blends of esters with mineral oils look much more promising.

Of course, for a synthetic lubricant to supplant a petroleum lubricant, the synthetic must be at least as good in most properties and very much better in one or more important aspects. Choice of lubricant must be a compromise between performance and availability. Because of their availability, petroleum products will continue in use as long as they can satisfy performance demands.

The interest in synthetic lubricants for military aircraft is a direct result of the striking advances made in the design of high-speed, highly maneuverable aircraft for use in all climates. Operation at high altitudes and from arctic bases requires lubricants that can serve at low temperatures. At the same time, operation with high-rpm accessories and from tropical bases requires lubricants to be capable of operating at high temperatures. Bearing or surrounding temperatures may reach 250 or 300 F.

In many cases where weight and space considerations dictated design of accessories such as generators, alternators, and air compressors for operation at temperatures as high as the materials would stand, the lubricant was the critical material.

With the highly refined and dewaxed petroleum stock of narrow boiling range necessary for satisfactory low-temperature operation, high-temperature properties such as volatility and viscosity were unsatisfactory. Conversely, petroleum products giving satisfactory performance at 250 to 300 F thickened excessively at -67 F.

But investigation showed that certain synthetic lubricants were less sensitive to temperature.

Best of the synthetics studied have been the dibasic acids adipic acid, azelaic acid, and sebacic acid esterified completely by 2-ethyl, hexyl alcohol. Table 1 shows their physical properties; Table 2 and 3 show their potential as lubricants. The sebacate had already been used successfully as a vacuum pump oil.

Also investigated were a number of other synthetic lubricants such as polypropylene glycol ether, dimethyl polysiloxane (silicone), hexachlorobutadiene, and tributyl phosphate.

Addition of suitable additives to the diesters made their oxidation and rust-inhibiting characteristics comparable to those of the best petroleum oils of the same viscosity grade. Additives useful for petroleum oils are generally effective for diesters. The diesters plus additives thickened by a lithium soap form a smooth and buttery grease.

The properties of these synthetics enabled them to perform reliably under conditions in which comparable petroleum products proved inadequate.

* Paper "The Air Force Looks at Synthetic Lubricants" was presented at SAE Metropolitan Section on March 17, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.) Opinions expressed in this paper are those of the authors and do not necessarily represent the official opinions of the US Air Force or the AMC.

Table 1—Physical Properties of Diesters[a]

	Di-(2 Ethyl-Hexyl) Adipate	Di-(2 Ethyl-Hexyl) Azelate	Di-(2 Ethyl-Hexyl) Sebacate
Molecular Weight	370.6	432.6	426.7
Boiling Point, F	385 at 2 mm	451-455 at 2.5 mm	450 at 0.5 mm
Freezing Point, F	-90	-108	-67
Density at 4 C	0.922	0.915	0.912
Water Solubility, % water dissolved	0.08	0.06	0.06
Viscosity, centistokes			
at 210 F	2.38	3.06	3.32
at 100 F	8.22	11.4	12.6
at 0 F	102	156	187
at -40 F	807	1190	1410

[a] E. M. Bried, H. F. Kidder, C. M. Murphy, and W. A. Zisman: "Synthetic Lubricants from Branched Chain Diesters," Journal of Industrial and Engineering Chemistry, Vol. 39, No. 4, April, 1947, pp. 484-491.

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Lubricate Military Airplanes

BASED ON PAPER* BY

Bernard Rubin and Edward M. Glass

Materials Laboratory, Engineering Div.
Air Materiel Command

Therefore, the synthetics became the basis for two new specifications, AN-G-25 grease and AN-O-11 oil.

For Specification AN-G-25 grease, sebacate and azelate diester oils and polyglycol derivative base oils have been used thus far.

AN-G-25 is generally suitable for all ball, roller or needle bearings used in aircraft—including prelubricated bearings for lifetime service—which normally operate at below 250 F and which for short periods of time may operate at temperatures up to 300 F. It serves on gears and other types of sliding surfaces under moderate loads. However, the grease should be service-tested before recommendation for use in sliding friction applications.

This grease allows equipment to operate at -67 F. But its viscosity at very low temperatures requires increases in torque which must be taken into account in equipment design.

Since the grease does not contain an extreme-pressure or special antiwear additive, it is not the proper lubricant for use in screw mechanisms, gears, and linear sliders where high loading is encountered.

Two years of field experience have shown that this grease performs satisfactorily when used as recommended. Cold-weather tests in Alaska and at the Eglin Field Climatic Hanger have shown also that the grease perfoms satisfactorily under extreme cold weather conditions.

It is lubricating all the antifriction bearings of surface control systems. These amount to about 70 or 80% of all bearings used in the airframe proper and include pulley bearings, hinge bearings, rod end bearings, movable surface cabin control bearings, and related equipment.

The AN-G-3a grease previously used in these applications bled oil excessively, leaving soap residues and depriving the bearings of protection against moisture corrosion. Besides, tests showed that at temperatures above 125 F, the grease lost oil by evaporation of volatiles and dried out. This precluded use of AN-G-3 types in those airframe bearings near turbo-superchargers, exhaust sections, and tail cone areas of jet aircraft, where temperatures exceed 160 F.

Diester-oil AN-G-25 grease is used also in small bearings running at speeds up to 20,000 rpm in electric motors and instruments, in large slower-moving bearings such as the 30-in.-diameter bearing with ½-in. balls in the B-36 turret, in sliding friction applications such as actuating mechanisms and guides for flaps and other working joints, in gear boxes and on screws for some trim tab actuators,

Table 2—Tests Indicating the Lubricant Potential of Pure Diesters[b]

	Adipate	Azelate	Sebacate
Evaporation Test (Open dish -150 F), % loss			
After 48 hours	0.03	0.03	0.02
After 168 hours	0.09	0.05	0.05
Neutralization No. Increase	0.01	0.02	0.01
% Viscosity Increase	Nil	0.01	0.01
Viscosity Index (Kinematic)	121	146	154
Flammability			
Flash Point (C.O.C.), F	380	445	450
Fire Point (C.O.C.), F	430	470	495
Spontaneous Ignition Temperature, F	743	756	765
Oxygen Demand	16	23	18
Dynamic Oxidation			
% Viscosity Increase	36.4	14.0	11.9
% Neutralization No. Increase	25.5	8.4	9.9
% Weight Loss	2.0	4.8	0.0
Rusting Tests[c]			
Modified Turbine Tests	..	..	10
Static Drop (72 hr)	10	10	10
Humidity Cabinet (24 hr)	..	..	10
Foaming	Nil	Nil	Nil
Hydrolytic Stability (pH_4 to pH_{10} at 212 F for 168 hr)	Max. acidity = 0.05 mg KOH per gm ester	As adipate	As adipate

[b] E. M. Bried, H. F. Kidder, C. M. Murphy, W. A. Zisman: "Synthetic Lubricants from Branched Chain Diesters," Journal of Industrial and Engineering Chemistry, Vol. 39, No. 4, April, 1947, pp. 484-491.

[c] 10 indicates maximum rusting; 0 indicates no rusting.

Table 3—Wear Characteristics of Pure Diesters and Specification Products Measured on Shell Four-Ball Wear Tester[d]

Type of Wear		Adipate	Azelate	Sebacate	AN-O-6a	AN-O-11
Steel on Steel						
	1 kg	0.32	0.39	0.26	0.31	0.39
Load	10 kg	0.64	0.63	0.55	0.61	0.59
	40 kg	0.84	0.79	0.69	0.81	0.84
Steel on Bronze						
	1 kg	1.31	0.50	0.65	1.56	0.52
Load	10 kg	1.27	1.04	1.10	2.40	0.82
	40 kg	1.43	1.43	1.40	3.12	1.38

[d] M. R. Fenske: Progress Report under Contract NORD 7958, July 12, 1948.

and throughout such electronic equipment as the APQ-7 radar antenna.

AN-G-25 greases have succeeded where AN-G-3a greases liquified in working in lightly loaded gear applications such as flap and magneto gears and in high-speed gyro rotor bearings.

One manufacturer uses a polyglycol derivative base fluid as the oil component of his AN-G-25 grease. Available laboratory test data on the 10,000-rpm high-speed high-temperature tester and a gear test rig indicate that it is at least equal to the diester type.

The other Specification, AN-O-11, covers a light oil containing a diester plus additives. It is used primarily for instruments and electronic equipment, especially where volatility of AN-O-6 and AN-O-9 petroleum oils is objectionable.

The cabin air compressor of the P-84 operates at a shaft speed greater than 100,000 rpm and temperatures as high as 310 F. With AN-O-9 oil, very high consumption and premature failures occured. But AN-O-11 lasts throughout the 25 hours between inspections.

The low volatility of AN-O-11 has eliminated the complaints of oil fogging on lenses of optical instruments.

Although synthetic greases and light oils are now in service, synthetic engine oils have not progressed so far.

Single-cylinder tests were enough to discourage use of engine oils in which the bulk of the lubricant was a diester.

However, blends of mineral oils and diester, of mineral oil, diester, and polyester, and of mineral oil and polyester are extremely promising. They can be stabilized to oxidation, show good detergency, have pour points of -60 F and lower, and—with proper additives—show corrosion protection equal to that of a petroleum engine oil. These materials are currently undergoing test-stand evaluation in standard multicylinder aircraft engines.

A synthetic investigated earlier for use as a winter-grade lubricant showed characteristics favorable to cold-weather operation and engine cleanliness but failed to protect against corroding agents formed in the engine and moisture corrosion. This synthetic was a polyalkylene glycol derivative designated PPO-265.

With this lubricant, combustion blowby products centrifuged out, pilots reported engine roughness and evidence of valve sticking, and certain plasticized synthetic rubber seals tended to shrink. It was concluded after service test that PPO-265 was unsatisfactory as an aircraft engine oil.

Common Alloy Steels

Continued from page 53

Steels with 0.45 – 0.50 mean carbon content are used primarily for gears and other parts requiring fairly high hardness as well as strength and toughness. Such parts are usually oil-quenched and a minimum of 90% martensite in the as-quenched condition is desirable.

Hardenability Classification—0.45 – 0.50 Carbon

(a) Low Hardenability		(b) Medium Hardenability			(c) High Hardenability
5045	9747	2345	5147	8745	4150
5046	9763	3145	5150	8747	9850
5145		3150	8645	8750	
			8647	9445	
		4145	8650	9845	

Steels with 0.50 – 0.62 mean carbon content are used primarily for springs and hand tools. The hardenability necessary depends upon the thickness of the material and the quenching practice.

Hardenability Classification—0.50 – 0.62 Carbon

(a) Medium Hardenability		(b) High Hardenability
4068		8653
5150	9254	8655
5152	9255	8660
6150	9260	9262
8650	9261	

Steels with a 1.02 mean carbon content (50100, 51100, and 52100) are straight chromium electric-furnace steels used primarily for the races and balls or rollers of antifriction bearings. They are also used for other parts requiring high hardness and wear resistance. The compositions of the three steels are identical, except for a variation in chromium, with a corresponding variation in hardenability.

Hardenability Classification—1.02 Carbon

(a) Low Hardenability	(b) Medium Hardenability
51000	51100
	52100

Some of the SAE alloy steels, 4024, 4028, and 8641, are made resulphurized so as to give better machinability at a relatively high hardness.

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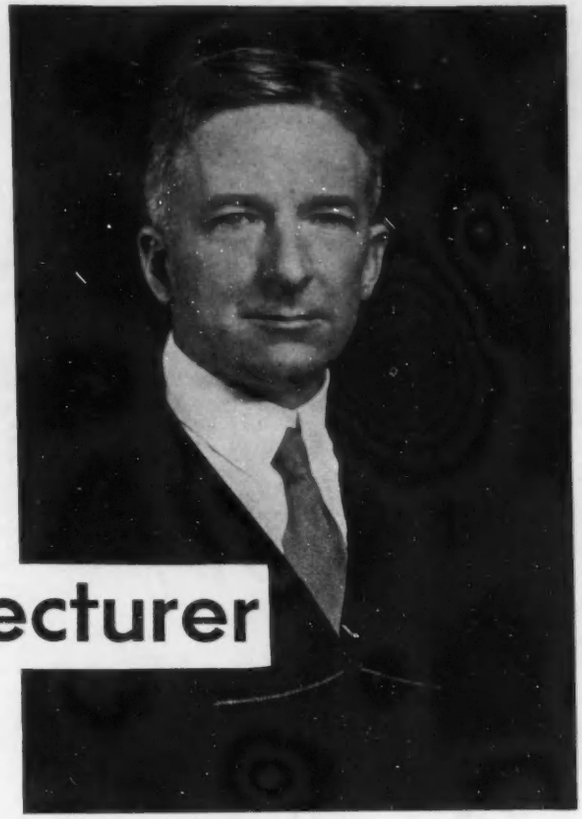
Chief Justice VANDERBILT N. J. Supreme Court



Chosen Beecroft Lecturer

CHIEF Justice Arthur T. Vanderbilt, of the Supreme Court of New Jersey, has been chosen the Third David Beecroft Memorial Lecturer for his major contributions to development of traffic laws and their enforcement. Justice Vanderbilt has rendered the "substantial contributions to safety of highway traffic" over a long period of years during which he has pointed the way, both by precept and judicial actions, to codes of traffic law and effective, undeviating enforcement procedures.

Justice Vanderbilt will deliver his Beecroft Memorial Lecture at the Statler Hotel, New York, Thursday evening, Oct. 20, at a dinner meeting under the auspices of the SAE Metropolitan Section.

Since 1938, Justice Vanderbilt has been chairman of the National Committee on Traffic Law Enforcement. He is also chairman of the advisory committee appointed by the United States Supreme Court to draft rules of criminal procedures for Federal Courts. Until recently, he was Dean of the New York University Law School. He was born in Newark, N. J., received his bachelor and master of arts degree at Wesleyan University, Middletown, Conn., and a graduate degree in law from Columbia University. Numerous doctorates have been conferred upon him in this country and in Canada.

Selection of Justice Vanderbilt was made by the SAE David Beecroft Traffic Engineering Committee, of which Pyke Johnson is chairman and SAE representative. Other members of the committee, each named by the organization which he represents, are: Lee C. Richardson, American Association of Motor Vehicle Administrators; R. H. Baldock, American Association of State Highway Officials; J. H. Hunt, Automobile Manufacturers Association; C. Donald Kennedy, Automotive Safety Foundation; Franklin M. Kreml, International Association of Chiefs of Police; and H. S. Fairbank, Public Works Administration.

Chairman of the SAE Metropolitan Section's Committee on Arrangements for the dinner is William E. Conway. Other members: Richard C. Long, Herbert Happersberg, T. C. Smith, P. E. Tobin, and O. V. Hinterther. The dinner is scheduled to begin at 6:30 p.m. and tickets are $7.00. Tables will accommodate 10.

The late David Beecroft as he appeared in 1921 when he was president of the Society of Automotive Engineers

The Beecroft Memorial Lectures are the result of a $2500 bequest to the Society by Past-President David Beecroft.

The money was to be used, Beecroft's will specified, "for ten awards of $250 each to be awarded at the sole discretion of the Past-Presidents Committee with the hope on my part, however, that the awards will be made for meritorious contributions to the improvement of traffic conditions. . . ."

The Council, on June 5, 1946, approved the Beecroft Memorial Lectures series recommended by its Past-Presidents Committee as the method of administering the bequest.

BASED ON PAPER* BY

L. F. Small

Lima-Hamilton Corp.

FREE-PISTON

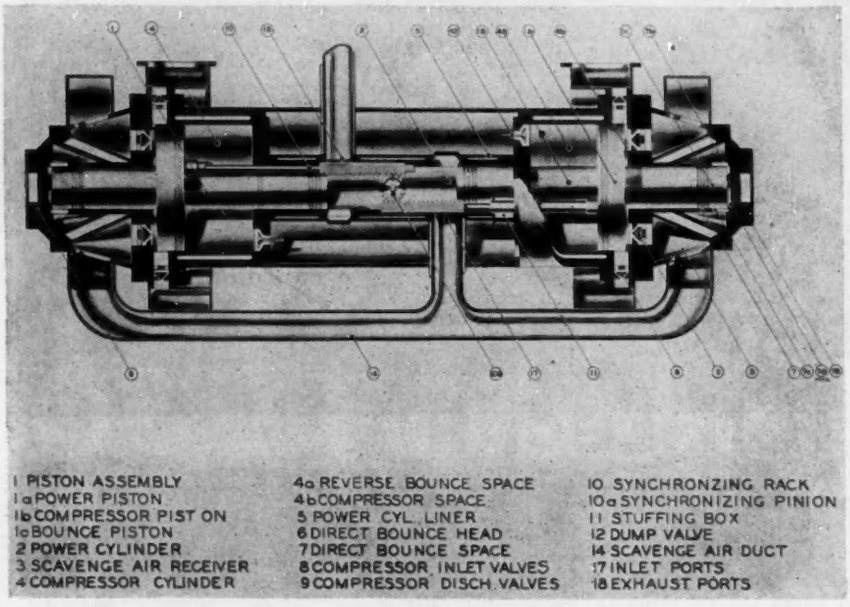


Fig. 1—Schematic diagram of gasifier

THE combination of a diesel cylinder using all its power output to supercharge itself and a gas turbine transforming energy of the hot compressed gases exhausted from the diesel cylinder into shaft power is being exploited here and abroad.

This diesel cylinder contains a pair of opposed "free" pistons which fly to their outward positions under force of combustion pressures, storing energy in the form of compressed air for their return stroke and compressing scavenge and charge air for the cylinder's next cycle. Except for the inevitable losses, all the energy developed from combustion is divided between the work of compression and the energy of gases leaving the cylinder. There is no flywheel.

Worthy partner of this diesel gas generator is the gas turbine, which provides a light-weight, compact means of expanding the low-pressure gases to atmospheric pressure.

What makes the combination of diesel cylinder and turbine particularly attractive are its conservation of scarce high-temperature-resistant alloys for turbine blading and its high thermal efficiency.

The Lima-Hamilton Corp. is developing these powerplants for locomotives and Navy ships. In France, stationary, locomotive, and marine plants are being built under the guidance of the inventor of free-piston machinery, Marquis Raoul Pescara.

The free-piston gas generator—or Pescara gasifier—appears in Fig. 1. Each piston assembly consists of three pistons: a power piston, a compressor piston, and a bounce piston. On the inward stroke, the power pistons compress the charge. Compression ignition takes place, and the pistons begin their outward stroke. The compressor piston compresses air for scavenging the power cylinder and for the new charge. The bounce piston compresses air in the direct bounce space.

When the pistons progress far enough out to uncover the exhaust ports, the hot compressed gases discharge into a receiver and go on to the turbine.

When the cool, lower-pressure charge replaces the combustion gases, the pressure of the air stored in the bounce cylinder against the bounce piston drives the piston assembly on its inward stroke.

The reverse bounce space serves in governing the unit and starting it. In Fig. 1, racks attached to the compressor piston and a pinion attached to the frame drive and time the fuel-injection pumps and synchronize the piston assemblies. The fuel nozzles are in the center radial plane of the power cylinder.

Figs. 2 and 3 show the gasifier with pistons at innermost and outermost points of their stroke, and the gas receiver and turbine. The figures also indicate the gas path.

Combination's Advantages

The gasifier takes the place of the compressor and combuster of the conventional gas turbine powerplant configuration. The gasifier has one big advantage over the compressor it replaces: It takes no power from the turbine. The axial or centrifugal compressor commonly takes about twice as much power from the turbine as the shaft gets. With the

* Paper "Free Piston Gas Generator Brightens Gas Turbine Future" was presented at SAE Summer Meeting, French Lick, Ind., June 7, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Dept. Price 25¢ to members, 50¢ to nonmembers.)

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GASIFIER
Conserves Turbine Alloys

gasifier, a given shaft power output can be achieved with a much smaller turbine.

The importance of the smaller turbine is that its blading takes less of the precious high-temperature-resistant alloys—which are becoming more expensive and harder to get as they improve.

Quantities of these alloys are definitely limited. Every indication points to the national necessity of minimizing competition with the aircraft powerplant industry in their procurement. Combining the turbine with the gasifier is one effective way to do it.

The gasifier-turbine combination has performance advantages too. As in all diesels, combustion occurs at high pressures and temperatures with very high heat releases. Yet the diesel is so well developed that there is no maintenance problem comparable to the periodic replacement of turboprop and turbojet combustors.

Secret of the diesel combustion chamber's success is its intermittent operation. Combustion is followed by cooling through expansion of combustion gases, then by scavenging, and finally by the new charge. Besides, cooling water surrounds the outer walls of the combustion chamber.

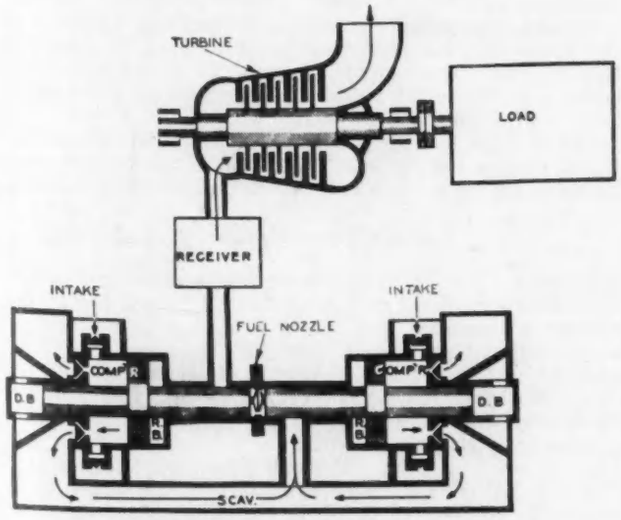
All gasifiers built so far use the opposed-piston type of construction, which gives the advantages of uniflow scavenging, perfect balance of reciprocating masses, and minimum cooled wall surface.

The peculiarities of free-piston motion contribute to the efficiency of the gasifier-turbine combination. Frequency of the gasifier—and consequently gasifier output and turbine shaft power—increases with decrease in piston weight. And the lighter the piston, the greater its acceleration under impact of combustion pressures, the earlier and more rapid the gas-temperature drop, and the smaller the heat loss to the cooling medium. All this adds up to better thermal efficiency.

The gas turbine expands the gases to atmospheric pressure, thus recovering some of the energy that escapes with the exhaust in conventional diesel engines. Because the turbine runs at high rpm's it is a light-weight and compact means for expanding the low-pressure gases.

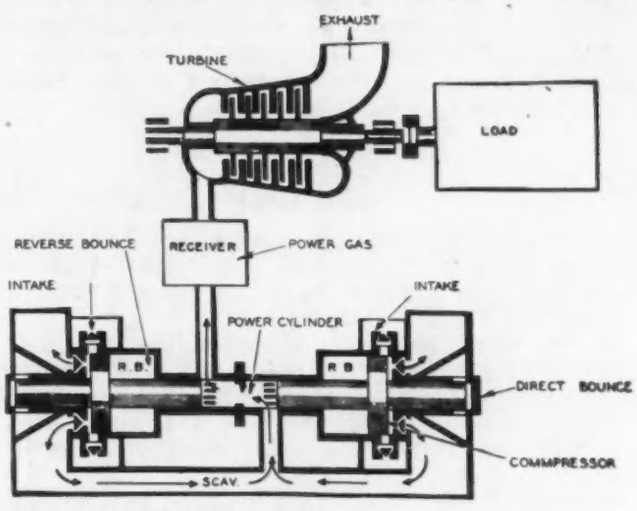
Lima-Hamilton is developing its locomotive and Naval-propulsion powerplants because the company is convinced that the gasifier-turbine combination is the most efficient way of producing power by combustion, as well as being light, compact, and reliable. The company believes the combination will prove competitive in outputs from 1500-20,000 hp, yielding to conventional diesels below and steam turbines above that range.

Lima-Hamilton's locomotive plant is now under construction.



INJECTION AND COMBUSTION

Fig. 2—Diagram of gasifier and turbine with pistons at innermost position



EXHAUST AND SCAVENGING

FREE PISTON DIESEL - TURBINE UNIT DIAGRAM

Fig. 3—Diagram of gasifier and turbine with pistons at outermost position

In France, SIGMA (Societe Industrielle Generale de Mecanique Appliquee) of Lyons builds free-piston gasifiers—and air compressors too; the Als-Thom works of Belfort builds gas turbines; and the Bureau Technique Pescara supplies technical and design guidance for gasifier-turbine combinations.

This French group has built many gasifiers of several designs, both vertical and horizontal, symmetric and asymetric in arrangement of compressors. Prof. G. Eichelberg of the Federal Institute of Technology in Zurich, who is consultant to the group, has reported one design in which a low-pressure bounce chamber receives all the physical energy of each combustion on the outward stroke. On the inward stroke, the energy is divided between the compressor and the compression in the power cylinder. This shortens the unit and makes it easier to maintain than the high-pressure bounce chambers of American design.

Some other entrants into the gasifier-turbine field have shelved their projects. Sulzer Bros. built a 7000-hp plant for merchant ship propulsion, tested it, then decided it could not compete commercially with the more conventional type of gas turbine powerplant. Pratt & Whitney Aircraft worked on a plant for aircraft propulsion, ran into difficulties with operation at atmospheric conditions existing above 20,000-ft altitude, and laid the project aside on that account. Several British developments were instituted but suspended also.

Most developments so far have been limited to single-cylinder gasifier units with ingenious arrangements in the blueprint stage for multiunit application. The next step is to perfect multi-cylinder gasifiers.

Proposals for large horsepower outputs made to date contemplate a number of single-cylinder gasifiers discharging into a few manifolds leading to one or more turbines. If Buchi turbo-charger practice can serve as giude, exhaust ducts serving two gasifiers timed to exhaust 180 deg apart should be used. Three or four pairs of such gasifiers properly timed with respect to each other should be mounted on a common frame and constitute one gasifier bank, with three or four ducts leading to a turbine.

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This list was compiled from the paper and discussions.

Excerpts from Discussion

A point worth mentioning is the noise level in the vicinity of the free-piston gas generator. The United States Naval Engineering Experiment Station at Annapolis has tested a German Junkers free-piston compressor and, with no muffler on the exhaust, the noise was somewhat objectionable to personnel in the vicinity of the test. How would a multiple installation of 10 or 15 units sound?

—John J. Dinan, USN Engineering Experiment Station

It has been my experience that the pecular free-piston motion is more a detriment to the design of all moving parts, including the compressor, than an advantage. Combustion—being a function of temperature, pressure, and time—must certainly be affected when the cylinder volume is changed so rapidly as to force the burning to take place even during the latter part of the working stroke.

It is important to differentiate between the efficiency of the gasifier and the efficiency of the gasifier-turbine combination. The efficiency of the gasifier is the percentage of the heat released by combustion of the fuel that is available to do work in the turbine. Therefore, 40% efficiency means that 40% of the heat of the fuel is available to do work in the turbine, but it must not be construed as being the efficiency at the end of the turbine shaft. A normal turbine efficiency would be approximately 85%. The overall plant efficiency is the product of the two.

—Ralph J. Hooker, American Locomotive Co. (Formerly of Pratt & Whitney Aircraft and Lima-Hamilton Corp.)

Pescara was a helicopter enthusiast and was continuously experimenting with full-size models. He needed a lightweight, efficient compressor to pursue his experiments but could find none on the market that suited his purpose. So he set about to develop a compressor, interrupting his helicopter developments. This new activity kept him busy for years until he was satisfied with the results of his free-piston compressor. He is now back at his helicopter development. But his free-piston engine is serving not only as a compressor alone but also in the gasifier-turbine combinations.

Factors contributing to the high thermal efficiency of the cycle are the relatively small amount of heat lost through the water jacket, namely 17 to 19%, and the absence of frictional losses due to connecting-rod angularity and crankshaft bearings.

The British use the rack-and-pinion arrangement for synchronizing the pistons. The French use the following bellcrank in some cases and pantograph motion in others.

—J. Barraja-Frauenfelder, Consulting Engineer

Actually, the Sulzer free-piston compressor, which was an experimental unit, though very heavy, proved extremely satisfactory. But Sulzer Bros. had under development at that time, besides the gas generator, two highly supercharged compound two-stroke diesel engines and the semiclosed-cycle marine gas turbine. All this development work was done without any government contract or subsidy, a sizeable program for a relatively small firm in a very small country. It proved impossible to get enough qualified engineers to do everything, and one of the three programs had to be abandoned. Although the free-piston machine program was dropped, the experience is still available to the firm.

—Prof. P. F. Martinuzzi, Cornell University

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Baltimore—Oct. 13

Engineers Club, Baltimore, Md.; dinner 7:00 p.m. Topic: The Berlin Airlift (lecture to be supported by actual films of this historic operation). Speaker: American Airlines representative, who will be announced later.

Central Illinois—Oct. 5 and Oct. 31

Oct. 5—Jefferson Hotel, Peoria, Ill.; dinner 6:30 p.m. Automobile Trends—R. J. Greenshields, director of research, Shell Oil Co., Inc., Wood River, Ill.

Oct. 31—Elks Club, Springfield, Ill.; dinner 6:30 p.m. Topic: Highway Planning and Design. Speaker: to be announced.

Chicago—Oct. 11

Knickerbocker Hotel, Chicago, Ill.; dinner 6:45 p.m. Meeting 8:00 p.m. My Friend, the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., and president, SAE. Social half-hour 6:15 to 6:45 p.m.

Mid-Continent—Oct. 14

Oklahoma A & M College, Stillwater, Okla.; dinner 6:30 p.m. Meeting 7:30 p.m. Meeting devoted to Diesel Activity. Speaker and subject to be announced.

Milwaukee—Oct. 7

Milwaukee Athletic Club; dinner 6:30 p.m. Air Cooled Engines for Vehicles—C. F. Bachle, vice-president in charge of research, Continental Aviation & Engineering Corp. Short movie and numerous slides. Subject will stress truck and bus application.

Detroit—Oct. 17 and 24

Oct. 17—Small Auditorium, Rackham Educational Memorial. Panel meeting 8:00 p.m. Subject: The Future of Radioactive Isotopes. Moderator: Roman B. Cuzack, mechanical designer, Ford Motor Co. Speakers: Ray James, research physicist, Chrysler Corp.; William Reid, research physicist, Ford Motor Co.; and Dr. Keith Symon, assistant professor of physics, Wayne University. Social hour will follow meeting.

Oct. 24—Small Auditorium, Rackham Educational Memorial. Panel meeting 7:30 p.m. Subject: Transfer Machines—Their Effect on Work Simplification and Cost Reduction. Moderator: Edward T. Ragsdale, general manufacturing manager, Buick Motor Division, GMC. Speakers: D. C. Burnham, manufacturing manager, Oldsmobile Division, GMC; O. R. Goodrich, supervisor of job evaluation, Packard Motor Car Co.; R. T. Hurley, president, Curtiss-Wright Corp.; and A. W. Zimmer, works manager, Reo Motors Inc.

Philadelphia—Oct. 12

Engineers' Club, Philadelphia, Pa.; dinner 6:30 p.m. Mechanical Octane Numbers—Alex Taub, Alex Taub Associates, Washington, D. C. Chairman: J. J. Mikita, vice-chairman, Fuels & Lubricants Activity, Philadelphia Section.

St. Louis—Oct. 11

Medart's; dinner 7:00 p.m. Subject: Fleet Maintenance (will discuss Selection of Equipment, Procurement, Operating Practices, Administration, Records and Reports, and Retirement of Equipment). Speaker: H. O. Mathews, manager of transportation, Standard Brands Inc., New York. Mathews, through years of experience in the operation of large fleets, has gathered a vast amount of knowledge in the operation of these fleets. This meeting should be of great interest to all fleet operators within this territory.

Southern California—Oct. 27

Nikabob Cafe, 875 S. Western Ave.; dinner 6:30 p.m. Meeting 8:00 p.m. Engineering Developments in Passenger Car Production—J. H. Booth, chief engineer, Thompson Products, Inc., Detroit, Mich.

Southern New England—Oct. 5 and Nov. 2

Oct. 5—Hartford Golf Club, West Hartford, Conn.; dinner 6:30 p.m. Airplanes Capable of Slow Flight—Prof. Otto Koppen, Massachusetts Institute of Technology.

Nov. 2—Sheraton Hotel, Springfield, Mass.; dinner 6:30 p.m. Locomotive Propulsion, Gas Turbine vs. Diesel—P. S. Vaughan, American Locomotive Co. Inspection trip of Springfield Armory will precede dinner and meeting, starting about 3.30 p.m.

NATIONAL MEETINGS

MEETING	DATE	HOTEL
AERONAUTIC and Aircraft Engineering Display	Oct. 5-8	Biltmore, Los Angeles
DIESEL ENGINE	Nov. 1-2	Chase, St. Louis, Mo.
FUELS & LUBRICANTS	Nov. 3-4	Chase, St. Louis, Mo.
ANNUAL MEETING and Engineering Display	Jan. 9-13, 1950	Book-Cadillac, Detroit
PASSENGER CAR, BODY, and PRODUCTION	March 14-16	Book-Cadillac, Detroit
AERONAUTIC and Aircraft Engineering Display	April 17-19	Statler, New York
SUMMER	June 4-9	French Lick Springs, French Lick, Ind.

Meeting participants took speakers to task in a question-and-answer session that closed the meeting. Facing the firing line are (left to right) A. T. Colwell, O. L.

ENGINEERS SCREEN

At West

MANIFOLD ways of bettering truck and bus performance and life—antidetonant injection, supercharging, aircooled engines, synchronized transmissions, bimetallic brakes, and light-weight trailers —were suggested at the SAE National West Coast Meeting, Portland, Oregon, Aug. 15 to 17. Each proposal ran into the cost challenge from economy-minded fleet men, some passing the "dollar" test, others revealing need for further development to bring cost in line with gains.

This highly successful meeting, sponsored by the eight West Coast Sections and Groups and the Diesel Engine, Fuels & Lubricants, Transportation & Maintenance, and Truck & Bus Activities, drew over 250 members and guests, with attendance high at individual sessions.

At the banquet, over 300 heard G. A. Bowie, Firestone Tire & Rubber Co., expound the importance of the individual in a talk "Under the Hood." He argued that we are all ignorant, but only on different subjects, and that from consideration and understanding of our fellow man stem our greatest achievements.

In technical sessions, commercial vehicle engineers and operators scrutinized suggested improvements in order of their appearance in the vehicle's power conversion and transmission routes.

Starting with engine fuel wants, engineers at the meeting learned that maximum antiknock requirements of commercial vehicles and passenger cars are nearly alike. This came out of a report on a nationwide test of 22 buses and 80 trucks running on a wide variety of fuels. These tests also showed that proper ignition system maintenance and adjustment will reduce the large variation in antiknock requirement among engines of the same make and model.

Comparison of these test results with those of a CRC survey made on 265 passenger cars (see SAE Journal, September, 1949, pp. 49–51, "Find Octane Needs of Cars on Road") showed a small difference between top needs of both types—about two octane units higher for cars over most of the requirement range. At 2250 rpm, it was shown that the difference between the CRC and commercial vehicle survey is larger, with car antiknock needs higher at all percentages of vehicles knock-free.

Another interesting item noted by the survey makers is the deviation of ignition timing from recommended specifications of the 102 vehicles in their as-found condition. Of the group, 49 were within a ± 2-deg range of the recommended setting, while 37 were retarded and 16 advanced more than this amount.

Operators were cautioned against neglecting distributor maintenance to keep down vehicle octane needs. Another way of lowering the vehicle's octane number requirement suggested at the meeting is antidetonant injection. This method was shown to have proved itself in the field, using the Vitameter to inject the antidetonant fluid.

For example, vehicles in the Yellow Cab Co. (Cleveland) fleet were put on straight-run gasoline, with the antidetonant Vitane (consisting of 85% methyl alcohol, 15% water, 3 cc lead, and an inhibitor) to make up the deficit in antiknock quality. Injected at a top rate of 15% of gasoline by volume, Vitane added 19 octane numbers to the 58 octane number gasoline. In this fleet's airport limousines, the gain was 25 octane numbers with a 22.6% maxi-

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Maag, Thomas Backus, F. C. Anderson, V. H. Stewart, James Myers, N. M. Reiners, D. D. Robertson, C. E. Stevens, Jr., Norman Hoertz, and H. J. Gibson

COST IDEAS

Coast Meeting

mum injection. Total fleet use of Vitane was about 4% of the gasoline used.

This and other fleet antidetonant tests were claimed to indicate no increase in engine wear, no detrimental effect on oil or lubrication, no increase in warm-up time, and no effect on starting. Some felt use of a straight-run gasoline, made possible by antidetonant injection, keeps engines cleaner and extends life to overhaul. It also should better oil consumption.

The fuel's combustion partner, air, also came in for a round of discussion. A specialist told fleet men that air cleaners for diesel engines should be suited to the engine and its operating conditions for economical operation. (For further details on air cleaner selection, see article "Application Defines Diesel Air Cleaner," p. 87.)

Supercharging Applications

Pumping more air into the engine to supercharge it is justified under only these three situations, engine men and operators agreed:

1. To help engines that are inherently bad breathers.

2. To compensate for loss in air density with increase in altitude.

3. To produce more power by introducing more pounds of air to burn more fuel.

Despite the first recommendation, supercharging to rectify poor design was said to be courting disaster. Designers urged cleaning up of poor air flow and combustion characteristics before resorting to supercharging. But they also noted that in most engines the cream had been skimmed off improvements in mechanical efficiency and the combustion process.

For operation at higher-than-sea level altitudes, the engine encounters less dense air so that its air poundage intake goes down. Available air decreases at the rate of 3% per 1000 ft, and horsepower decreases proportionately. In this case the supercharger is geared to pump enough volume of the less dense air to equal the same pounds the engine would consume with natural breathing at sea level.

In supercharging for greater power output, engine men recognized the pitfall of consuming all the added power in driving the supercharger. To keep horsepower loss in driving the supercharger to a minimum, the supercharger and engine should be carefully tailored to suit actual air requirements, they advised.

Cold-weather starting is no more difficult with a supercharged diesel than with a naturally aspirated engine, discussions on operations brought out. One company got very satisfactory starts at temperatures as low as −48 F by using a flame thrower. This consists of injecting the fuel into the manifold and igniting it with a spark plug to raise the temperature of the intake air. Ether also has been used as a starting aid.

The switch to engines found interest focused on aircooled powerplants, claimed as a feasible replacement for the cast-iron liquid-cooled types from engineering, performance, and economic viewpoints, with handsome gains to be made.

First big advantage of the aircooled engine for motor vehicles is complete freedom from plumbing difficulties and antifreeze troubles. Little trouble, if any, is anticipated from air-cooling fins, as mili-

Left to right: J. W. Sinclair, chairman of the 1950 SAE National West Coast Meeting, SAE President S. W. Sparrow, and Z. C. R. Hansen, chairman of the 1949 SAE National West Coast Meeting

tary wartime experience with aircooled engines proved.

Complete elimination of the cooling system, aircooled proponents continued, cuts engine weight to from one-third to one-half that of the liquid-cooled powerplant. Eliminating the radiator and cooling system also saves space. Using an aircooled instead of a liquid-cooled engine in one military vehicle shortened its length by 26 in.

All agreed that the aircooled engine is noiser because it does not have the water and extra metal density of the liquid-cooled engine to suppress sound. But some felt that adequate noise reduction could be achieved through design and sound insulation.

Ways to Valve Longevity

Engineers at the meeting took another look into the engine and pulled out the plum of longer-lasting exhaust valves through wise choice of valve material, design, and installation methods.

Exhaust valve materials were classified into three categories—ferritic, austenitic, and sigma precipitate groups. While the first material is easy to mill and machine, it does not have the resistance to fatigue and corrosion of the other two types. But the latter materials are tough to manufacture.

Chief suggestion made here is that the valve steel should suit the particular engine. In any given engine, one steel may perform better than another. Metallurgy discussions came around to truck fuel attack on bronzze valve guides. Steel rings on top of the guide was said to solve this.

Valve design features that guard against premature failures—such as burning and cracking—are hard facing and sodium cooling. While hard-faced valves were said to extend valve performance, they were not considered a cure-all. Sometimes the valve material is severely burned behind the hard facing due to cracks in the facing. High thermal and mechanical stresses in the ring of hard-facing material cause this. Sodium-cooled valves supress such action by heat-transfer action of the sodium, engineers were told. They often lower octane requirements by two to three units.

Another cause of recent valve troubles was traced to new compounded oils, which leave deposits on valves and/or valve seats. This deposit breaks down in sections and causes valve burning and warping. Valve rotators emerged as the remedy for this trouble.

In addition to selecting proper valve material and design for the engine, proper installation was advised for long valve life. For example, concentricity of valve seat with valve guide to within 0.002 was urged. An interference angle between valve seat and valve face with contact point at top also was recommended.

Perusal of suggested improvements continued via the power conversion and transmission paths, stopping next at the transmission, where engineers showed why the torque converter wouldn't work on trucks, except in special cases. But they did see gains using a synchronized auxiliary with widely-spaced steps in combination with a main transmission having closely-spaced ratios.

Two kinds of losses were shown to damn the torque converter for truck use. First is poor efficiency of the converter, the price paid for an infinite range of ratios. Second, several hundred rpm's are lost before dropping into the torque multiplying range. Thus, whatever drop in rpm from peak running speed is required to go into the conversion range must be added to the efficiency loss in comparing an hydraulic drive with mechanical conversion of torque.

However, several advantages for the hydraulic drive were pointed up . . . improved maneuverability and less driver effort. Some had a good word for the torque converter in multi-start operations. It was claimed that this would materially reduce maintenance since the clutch no longer has to pick up the load from a standing start and fewer shifts are necessary.

Discussions of more suitable truck transmissions brought forth the suggestion of a front transmission with closely-spaced ratios combined with an auxiliary with two or three widely-spaced ratios, each covering a range equal to one step more than the entire range covered by the front transmission. The auxiliary might have a starting range, a mountain driving range, and a high-speed range. Claimed for such an arrangement was the great reduction in the number of double shifts necessary.

The power-shifted synchronized auxiliary, transmission men continued, makes possible the use of such a combination by an unskilled driver. The shift between ranges is progressive and takes place automatically after preselection by the driver. It

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is no more difficult to make any one of the closely-spaced steps in the main transmission, they said.

Synchronized truck transmissions received other boosts from designers who said they were safer, easier to drive, and faster and more dependable in shifting than conventional types.

Shifting speed advantage was illustrated by comparison of the operational sequence using the conventional and synchronized drives. The conventional transmission involves these six steps in a double clutch shift: (1) declutch, cut throttle, (2) shift to neutral, (3) reclutch, (4) declutch, (5) shift into fifth, and (6) reclutch, apply throttle. The synchronized transmission halves the number of steps in this operation, with the sequence as follows: (1) declutch, (2) move shift lever directly toward fiifth speed position, applying pressure until shift is completed, and (3) reclutch, apply throttle.

On the safety angle, the synchronized drive was claimed to be better when shifting down to lower gears on steep grades for engine braking. Missing such a shift may result in a runaway and a serious wreck. It is possible to delay such a shift much longer with the synchronized transmission than with the conventional constant mesh transmission.

Discussions followed the power transmission train to the wheel, where attention centered on a bimetallic brake drum—aluminum and iron—said to dissipate large amounts of energy without exceeding a temperature at which linings lose their effectiveness. It substitutes a heat exchanger for a heat reservoir.

Based on an aircraft engine cylinder barrel development, this drum consists of a finned external aluminum barrel bonded to an iron liner. Test results proved that a molecular bond exists between the aluminum and ferrous members. Better heat conductivity of the aluminum makes possible the heat dissipation. The fins give 10 times the cooling surface presented by a conventional smooth drum. No heat checks were found on the bimetallic drums, except in areas of poor bond between liner and structure.

In a midget racing car, the bimetallic brakes showed two advantages over conventional brake drums. First, the brakes did not fade after a few applications, as with conventional drums. Second, the brake lasted a whole racing season rather than just one week.

Another test program verified the fast recovery of bimetallic brakes, which permits the brake to start its braking at a lower temperature and not build up to impossible levels with continued use.

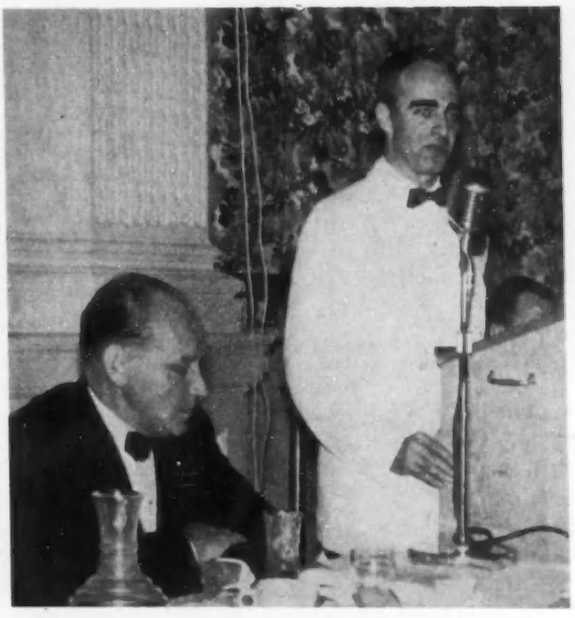
On the debit side of the bimetallic brake picture it was reported that additional air must be directed at the fins for full effectiveness. One way suggested was the use of deflecting scoops to impinge on the fins. For more severe braking, such as heavy truck rigs, it was felt that fractional horsepower blowers could prove economically feasible.

Use of aluminum and other light-weight metals also manifested itself in discussions of economical trailer design. Fleet men seeking top payload-carrying ability in trailers coupled with safety and strength were urged to use stainless steel as the basic material. They also were told to examine other materials—such as aluminum, magnesium, carbon steel, and alloy steel—so that the best may be chosen for each function.

Detailed examination of component designs gave

John G. Holmstrom, toastmaster (left), with dinner speaker George A. Bowie

J. S. Poulson, chairman of the SAE Oregon Section, opened the dinner program



way to discussion of the vehicle in its entirety, with preventive maintenance emerging as a salient factor in operational economy. Engineers heard from two fleet men about the Los Angeles Department of Water and Power's procedure schedule for preventive maintenance that pays off.

This preventive maintenance and inspection procedure is divided into five schedules, ranging from daily maintenance service to extensive teardown and repairs. The program relies heavily on engine oil analysis and dynamometer diagnosis, said these Los Angeles Fleet operators, which pay dividends by uncovering faulty conditions before they become serious.

Technical appraisal of these design proposals balanced against the dollar-and-cents side of the equation gave the real clue to their practicability.

For example, it was said that early Vitameter designs proved uneconomical because of high antidetonant consumption. Vitane was supplied at low intake manifold vacuums as well as at high speed, wasting the fluid in the latter situation. At high engine speed, antidetonant usually is not needed because of the normal drop in engine octane requirement.

But a more recent development described makes the Vitameter cut off the Vitane at a high engine speed, appreciably reducing the amount of fluid used. With this type of device, it was felt that antidetonant injection may well be economically attractive. Particularly is this so where improved fuel economy and power output can be achieved by advancing the spark timing. Instead of going to a higher quality fuel to satisfy the greater octane need of the advanced spark, antidetonant injection could supply the added engine antiknock wants.

No higher first costs and perhaps greater revenues were seen for the switch to aircooled engines in commercial vehicle operation. In fact, at high yearly production rates, the aircooled engine may be cheaper to build.

For one, it was pointed out, that many of the parts of both aircooled and liquid-cooled engines are alike. Using aluminum for the cylinder head and crankcase instead of cast iron for the aircooled design does not cost more since less aluminum by weight is used. Although on a volume basis aluminum would cost about 25% more than cast iron, this would be partially offset by some reduction in metal found in recent designs. Additionally, aluminum machines much easier and lends itself to cost-saving automatic processes such as permanent mold die casting.

Because the aircooled engine weighs less, additional payload could be carried. An aircooled engine for a bus recently built saves 1600 lb, equivalent to 10 passengers. This weight-saving was estimated to be worth $800 to $1600 per year in revenue, assuming capacity operation.

Investment in longer-lasting exhaust valves and air cleaners met with general approval from fleet men who felt the additional cost could save countless maintenance dollars later. Premature failures and road breakdowns stemming from inadequate parts cost the operator many times the difference in price between really long-service and run-of-the-mine components.

This argument proved equally potent in transmission discussions, where engineers agreed that the synchronized drive costs more initially, but less to maintain. It was borne out by a survey of 16 fleets, operating a total of more than 2000 trucks. Reporting on this survey, one engineer said that synchronized units ran an average of twice the mileage between overhauls.

Economics of torque converter operation proved less encouraging. The lower efficiency, especially at high torque multiplication ratios, means fewer miles per fuel dollar to fleet men. And this simplified shifting drive adds about 700 lb to the vehicle, they argued, in addition to which the torque converter costs much more than geared units. West Coast truckers, shooting for higher payloads via weight-reducing alloys, did not take kindly toward a potential loss in revenue due to the heavier converter.

But they enthusiastically received word about lighter metals in trailers that increase pay-load carrying ability.

For example, a stainless steel trailer was said to net up to 29 cu ft of additional capacity and a 2095-lb saving over a conventional trailer in the 35-ft class. In floor materials, magnesium alloy offers an 800-lb saving in a typical 35-ft dry cargo van and as much as 1250 lb in a refrigerated unit. Forged aluminum wheels, trailer designers said, are actually stronger than cast steel wheels and weigh 93½ lb less per wheel with 20-in. tires.

The aluminum-iron bimetallic brake drums can be permanent mold cast in large quantities at a price not much higher than that of present high quality drums, cost inquiries on this development brought out. The finned configuration on large-size drums for commercial vehicles has been successfully sand cast, and should not command too high a premium price since the fins will not be machined.

Under the general chairmanship of **Z. C. R. Hansen**, the following served as chairman of the seven technical sessions of the SAE National West Coast Meeting: **David Brown, W. B. Keith, R. W. Moore, R. A. Hundley, Harold Puxon, R. E. Fleischer,** and **J. W. Sinclair.** This report is based on discussions and 12 papers . . . "Antidetonant Injection from the Fleet Operator's Viewpoint," by **A. T. Colwell,** Thompson Products Inc.; "Lubrication of Heavy Duty Roller Applications," by **O. L. Maag,** Timken Roller Bearing Co.; "Transmissions and Engine to the Road Gear Ratios," by **Thomas Backus,** Fuller Mfg. Co.; "Modern Maintenance and Preventive Maintenance Procedures," by **F. C. Anderson** and **E. W. Templin,** Los Angeles Department of Water and Power; "Aircooled Engines for Vehicles," by **C. F. Bachle,** Continental Aviation & Engineering Corp.; "How Would You Design a Trailer for Maximum Efficiency?" by **L. H. Chaillé** and **V. H. Stewart,** Freuhauf Trailer Co.

"Air Cleaners for Diesel Engines," presented by **James Myers,** Farr Co.; "Why Supercharge?" by **N. M. Reiners,** Cummins Engine Co.; "Synchronized versus Conventional Transmissions in Truck Operation," by **D. D. Robertson,** Spicer Mfg. Division, Dana Corp.; "Design and Development Considerations of a Bimetallic Brake Drum," by **C. E. Stevens, Jr.,** Al-Fin Division, Fairchild Engine and Aircraft Corp.; "Recent Trends in Valve Design and Maintenance for Automotive and Diesel Engines," by **Norman Hoertz** and **R. M. Rogers,** Thompson Products, Inc.; and "Antiknock Requirements of Commercial Vehicles," by **H. J. Gibson,** Ethyl Corp.; **G. W. Pusack,** Socony Vacuum Oil Co.; **J. A. Edgar,** Shell Oil Co., Inc.; and **R. J. Greenshields,** Shell Oil Co., Inc.

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—Predicts Design Trend For High Compression Ratio Engines

Based on paper by

ROBERT P. ERNEST

Kaiser-Frazer Corp.

THE valve-in-head design and "V" cylinder head arrangement will predominate in coming high compression engines because of the engineering advantages they offer.

Of the four valve arrangements tried in this country, as shown in Fig. 1, the "I" or valve-in-head is best suited for the high compression ratio job for three reasons: (1) it has high volumetric efficiency at high compression ratios, (2) its thermal efficiencies are higher due to the small area of combustion chamber exposed to the coolant, and (3) it is easy to maintain and service.

Although the valve-in-head type engine presently powers only seven of the 30 American cars, a high percentage of automotive engines will be of this type within the next five years.

The L-head engine, currently found in the other 23 American cars, is least expensive to make, but has acceptable breathing only up to a compression ratio of 8 to 1. The F-head and T-head engines both are limited as to compression ratio and have a large surface of combustion chamber exposed to the coolant.

On number and arrangement of cylinders, sixes and eights power all American cars, with the exception of the 4-cyl Crosley. Straight eights are losing favor with engineers, particularly for high compression designs, because of their objectionable weight and length, and the problems they present in manifolding and crankshaft design. The V-type engine appears to be replacing the straight eight.

The V-8 with overhead valves offers many attractive design features to the engine designer. A stroke-to-bore ratio of less than one, which reduces piston speed, is desirable for this type of cross-section. The short stroke increases the bearing overlap, which gives the crankshaft greater rigidity. The cylinder head design allows short exhaust ports, which reduces the heat rejection to the coolant and results in a smaller and lighter radiator core.

V-8 Production Costly

More complicated intake manifolding and more costly machining are the disadvantages of the V-8 type engine.

The line-6 engine has many inherent advantages, such as good distribution control, machining simplicity, and good accessibility for servicing. To overcome objections to the line-6 length, development of a V-6 is under way. Trouble with this arrangement is that to maintain even firing pulsations, there is a primary couple that cannot be balanced out without the aid of a separate balance shaft. If the V is 90 deg, the couple can be eliminated, but the firing becomes uneven. It seems that a well-designed motor mount would keep the unbalance from being objectionable.

If this assumption is true, the V-6 would be an ideal engine from space, weight, performance, manifolding, and cost standpoints.

Unpopularity of the 4-cyl engine is the fault of advertising, not performance of the engine. Despite the motor mount problem stemming from the unbalanced secondary inertia force and the need for a slightly heavier flywheel to smooth out torque pulsations (somewhat affecting engine acceleration qualities), the gains in friction, size, and cost are tremendous.

That the 4-cyl engine is less powerful than the six, and both of these less powerful than the 8-cyl powerplant is a misconception. Fact is that most racing events in this country are won by cars with 4-cyl engines.

Another must to make high-speed high-compression engines durable is block construction with adequate main bearings. High compression engines must have a main bearing between each crankpin journal. This minimizes main bearing loads, eliminates offset connecting rods, and allows a shorter block. It also reduces friction horsepower by lowering deflection, despite the extra bearings.

The trend also seems fairly clear as to performance of coming engines—torque peaks at higher speeds. With automatic transmissions becoming increasingly popular, the old-fashioned high torque at low speeds is bowing to a modern trend of high-output high-speed engines with peak torques at around 2000 rpm. (Paper "Engine Design," was presented at SAE Detroit Section, May 2, 1949. This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

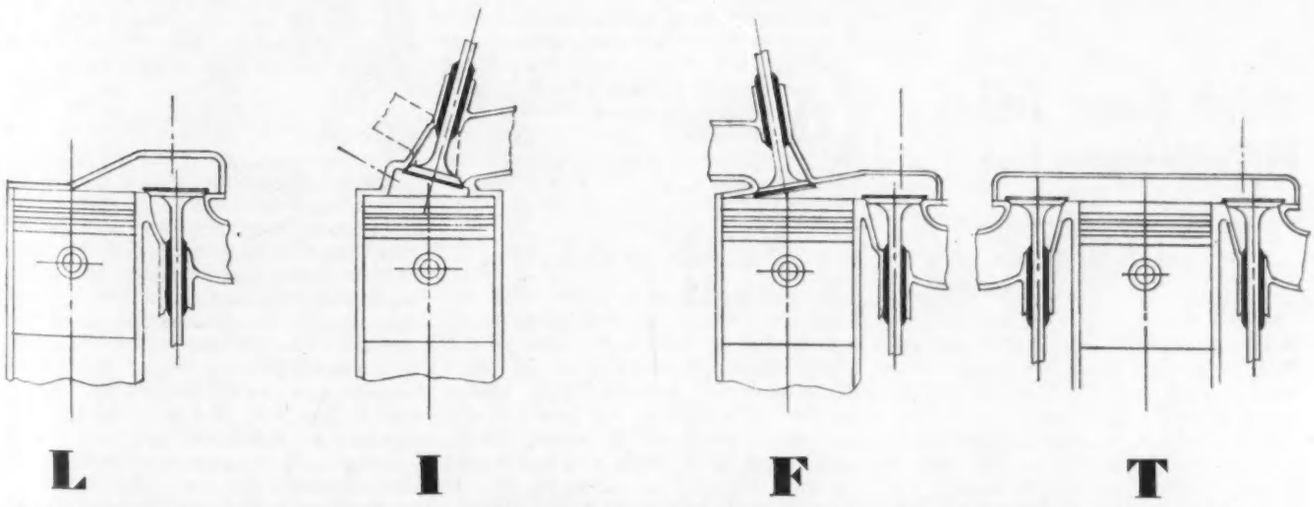


Fig. 1—Four types of valve arrangements are, left to right, the L-head, the I-head or valve-in-head, the F-head, and the T-head. Only the first two are found in production automotive engines today, and the valve-in-head appears to be the best for high compression engines

Basic Design Carried Through Engine Series

Based on paper by

L. F. SHOEMAKER

The Buda Co.

FROM one basic engine design Buda has developed a series of engines, ranging from an 8-cyl to a 1-cyl powerplant, that can be adapted to any type of fuel.

About 10 years ago the basic engine type was decided upon together with tool equipment. It was to be an 8-cyl, supercharged, liquid-cooled, in-line diesel. The supercharging determined crank and rod size as well as bearing dimensions. By shortening the engine, a 6, 4, 3, 2, or 1-cyl engine could be built from this basic structure. This made possible parts interchangeability.

Also it was possible to work out easily the naturally aspirated version of these diesels and to provide the engines with ignition systems and carburetors instead of injection pumps. Thus it became possible to burn satisfactorily fuels such as natural gas, butane, high or low octane gasolines, or even kerosene and other low grade tractor fuels.

This represents an effort to streamline the basic design of an engine series to meet as many of the potential customer requirements as possible. (Paper "Larger Diesel Engines for Rubber Tired Equipment," was presented at SAE Metropolitan Section, New York, April 21, 1949. This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Weight Saving Trends In Trailer Selection

Based on paper by

JAMES J. BLACK

Trailmobile Co.

CONTINUED effort by vehicle designers and fleet operators have reduced weights of trailers in the interest of larger payloads.

Basic design depends upon state gross weight limitations, but use of lighter lumber for floors, taking advantage of ribbed sheet metal, and using lighter alloys have proved advantageous to the operator.

For example, white oak weighs 4¼ lb per board foot on the average. Edge grain fir offers a saving of 25%. Alloy aluminum plate floors are finding increased favor, but they represent nailing problems.

In using fir one must keep in mind that its load carrying capacity is 90% that of oak, its shear is 60%, and denting about 50% that of the harder and heavier wood.

Ribbed sheets fastened to all verticals is a weight saver. Consideration must be given to shear in the panels and the number and size of rivets in relation to panel thickness.

Because of the low stresses in the panel of about 1500 psi, light alloys will be as efficient as heavier steel panels. In such structures the panels carry loads, and they must be maintained with care.

Corrugated sheeting is more generally used in the truss or stressed panel.

Axle Loading Regulations

Most states recognize axle spacings of over 4 ft to be no rougher on roads than single axles with the same load per axle, and allow increased payloads. Virginia gives the smallest increase for states recognizing tandems. There the ratio of payload to gross load favors the tandem by 5% without consideration to operators' wages, gasoline, or other expenses.

Sixteen different axle positions are found for each given length in the 48 state laws for rear axle locations on tandem semi-trailers. In those states where bridge formulas are used as the basis of loading, gain in overall weight can be made by lengthening the wheelbase and maintaining the same distribution.

A conventional 32 ft tandem van trailer weighs 11,200 lb. With aluminum panels it will weigh 10,500. By redesigning for light weight this can be reduced to 8500 lb.

Generally used insulating materials for bodies which will haul refrigerated products are fiber glass, fireproofed cotton, kapok, cellular rubber, and cork. Factors in choosing such materials include water absorption, weight, tendency to pack down, proof against vermin, and chemical stability.

New Insulating Materials

Newer materials now available include a foamed styrene weighing about 2 lb per cu ft, with a K factor of 0.3, and a loading capacity of 20 psi. This excludes water because of its sealed cellular construction. A new cellular sponge rubber has a K factor of 0.22, weighs five lb per cu ft, and has a load carrying capacity of about 40 psi. It is practically free from water absorption.

One of three means of refrigeration used for perishables is the use of wet ice in the load or in a wet ice bunker up forward. Air is circulated around the load by a centrifugal blower. Another is an auxiliary engine, compressor, and condenser mounted outside the vehicle with the evaporator of the system inside.

A third method, the Polo-trol system, uses dry ice as the primary coolant. This cools a secondary solution which is pumped into wall plates installed lengthwise in the trailer. Movement of the fluid is thermostatically controlled, and a safety system allows circulation by controlled syphon should the pump fail. (Paper "Commercial Trailer Selection" was presented at SAE National Transportation Meeting, Cleveland, March 29, 1949. This paper is available from SAE Special Publications Department. Price: 25¢ to SAE members, 50¢ to nonmembers.)

Photography Aids Wasp Development

Based on paper by

R. E. GORTON

Pratt & Whitney Aircraft

SPECIAL photographic techniques, both movie and still, played an important part in the instrumentation for developing the 28-cyl Wasp Major aircraft engine.

One type of camera used was an 8 mm movie camera, developed by Western Electric, capable of taking pictures at the rate of 8500 frames per sec. Projecting such movies at the normal rate of 16 frames per sec gives a time magnification of over 500 times. Such pictures permit the observation of many phenomena which cannot be seen at their actual rate of occurrence.

Fin Vibration Stopped

For example, one of the untried features developed for the Wasp Major engine was the cylinder head design. Because more cylinder cooling fin area was required than on previous engines, very thin and deep finning had to be used. On early cylinders of this type, vibration of the fins, excited by firing shock in the cylinder, caused the fins to break off at their bases. There was no ready way to attach vibration pickups to the fins, so the vibration was viewed by high-speed movies.

By taking close-up photographs of all critical fin areas at various engine conditions, it was readily possible to pick out points of maximum amplitude.

The high-speed movie camera also served in the study of a giant-size "tin

can"—a steel drum designed for shipping Wasp Major engines. About 5 ft in diameter and 11 ft long, this drum normally rested on end. There was the possibility that in handling it would be tipped over and fall without restraint onto its side.

To get some notion of the speeds and accelerations present at the time it hit the floor, high-speed movies were taken showing the falling drum and a stationary measuring scale. Analysis of the film, frame by frame, gave a plot of drum position versus time at 1/8000-sec intervals.

Problem of Light Intensity

This application is typical of the many non-recurring events of very short duration that must be caught by a movie camera that expends its 100-ft roll of film in about 1 sec. Automatic electrical synchronization of camera and event is essential, but at the same time a major problem with such instrumenation. Another difficulty is getting enough light for pictures at 8500 frames per sec. Sufficiently great intensity to give proper exposure often is accompanied by so much heat that paint or other combustible material is destroyed in a few seconds.

Photographing Fuel Spray

During development of fuel injection equipment used in the latest Wasp Major engines, high-speed movies of the spray from the injection nozzles, taken at an exposure time of about 1/25,000 sec, proved to be inadequate. While movies at this exposure showed the general pattern of the spray, they did not stop motion of the droplets to give a sharp picture.

For a finer detailed instantaneous picture of the spray droplets pattern, a "Microflash" light source was used in conjunction with an ordinary still camera. Such pictures, with an exposure of about 4/1,000,000 sec, did the job. Intensity of the flash from the mercury vapor discharge tube of this light source is sufficient for good exposures, even though the exposure is very short.

An allied instrumentation technique stems from use of a similar short-duration light source, made to give recurrent flashes at a rapid and uniform rate. By synchronizing the flashes with the speed of rotation or frequency of vibration of a member, the stroboscopic effect causes an apparent slowing or stopping of recurrent motion.

These stroboscopes may be flashed at an adjustable rate by means of an oscillator, in which case they may be used to measure speed or frequency. Or they may be triggered by an electrical impulse generated by some device on the moving or rotating member; here they serve apparently to stop its motion.

Such stroboscopes were used widely during development of Wasp Major components such as oil pumps, oil seals, accessory drives, valves, valve springs, and rocker arms. (Paper "Instrumentation as Applied to the Development of the Wasp Major Engine," was presented at SAE So. New England Section, Hartford, March 10, 1949.)

Decentralization Saves Truck Operator $42,400

Based on paper by

JOSEPH HUSSON

Consumers Co.

ABILITY to store trucks outside in Chicago's winter weather led Consumer Co. to change from centralized to decentralized truck operations. This netted a saving of $42,400 per year in 1948 as against an estimate of $45,000, with about half the decentralization program carried out.

To interpret these results properly, is necessary to know that prior to the

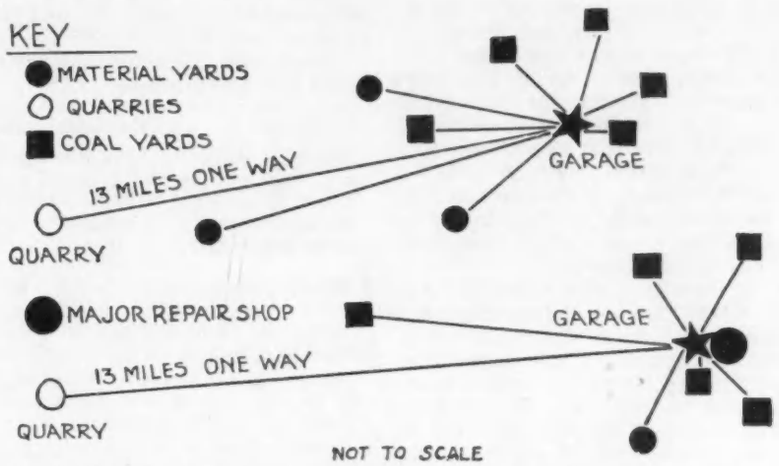


Fig. 1—Under the old plan, Consumer Co.'s trucks were dispatched from two centralized garages

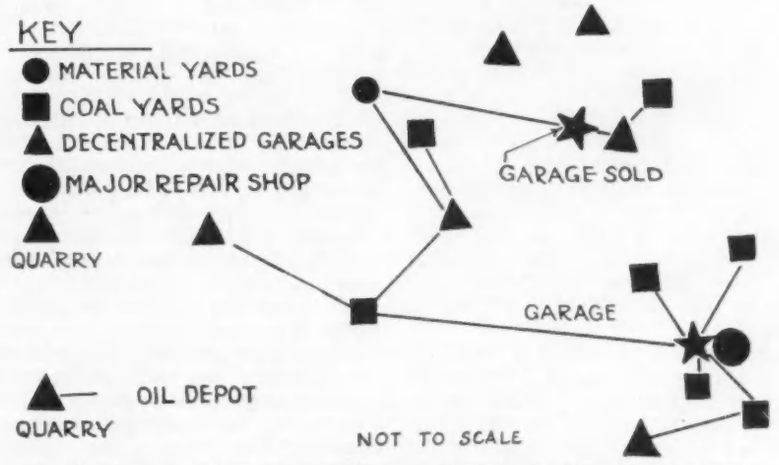


Fig. 2—Consumer Co.'s revised operation, dispatching trucks from several decentralized garages

fall and winter of 1948, the fleet of about 150 trucks was largely serviced and stored overnight inside of two centralized garages, one on the North Side and the other on the South Side.

Each morning the trucks were driven empty to two stone quarries, four city building material yards, ten coal yards, and three fuel oil yards from which deliveries of these products were made to customers in the Chicago area. Each night the trucks were returned to the two centralized garages for similar quarry or yard assignment on the following day.

The problem was to determine if some means could be provided to garage and service the trucks each night at the yards from which they worked during the day. In this way the company could save the expense of idle or nonproductive truck mileage and drivers' wages in running morning and night between the two centralized garages and the various yards.

Saving Versus Cost

It was further necessary to determine if the cost of providing the means to operate the trucks out of the local yards would exceed or be less than the value of idle mileage and driver hours saved. Thus the problem resolved itself into first finding out how many truck miles could be saved if all of the company trucks operated out of local yards and, second, how much would it cost to make decentralized operations possible.

A six-month survey showed that about $60,000 could be saved by garaging all the trucks in the yards out of which they worked. It was made up of $24,000 for an estimated 230,608 miles eliminated between the two centralized garages and the various yards at an average cost of 10.7¢ per mile, plus $36,000 for drivers' pay, made up of 18,592 hr at an average overtime rate of $1.88 per hr.

But subtracted from this $60,000 must be the precautions and facilities to store and service the trucks at the various yards. Precautions include the use of antifreeze and testing of the radiator cooling fluid at least once a day. Minimum facilities required are a heated one-stall building, large enough to house one truck so that it can be worked on during summer rains and winter snows; a workbench with adequate tools and service equipment; space for spare tires, drums of motor and hoist oil, and antifreeze; and facilities for storing and dispensing gasoline.

Further study indicated a need for 11 such one-stall buildings and two two-stall structures at which 20 or more trucks would be stored. This made a total of 13 new buildings for a net total of 17 locations, since two of the fuel oil yards were combined with coal yards. The difference between the 17 locations and the 13 planned new buildings contemplated the remodeling and use of existing heated building facilities at four yards.

Capital Expense

Total investment for this was estimated at $65,600 which amounts to about a $4000 yearly expense, based on a 20-year building depreciation and a 10-year equipment depreciation.

Additional increased expenses included $1000 per year for antifreeze not previously used, $3900 in increased mechanics' wages, $2900 for increased night use of regular day service trucks, and $32000 for delayed driver time in getting the trucks started on extremely cold mornings.

This increased yearly expense totals $15,000. Deducted from the gross savings of $60,000 leaves an indicated net yearly saving of $45,000. At this saving, the new capital investment required would be returnabie in 1.5 years.

Actual savings of $42,400 realized in 1948 involved only nine decentralized garages instead of 13 originally contemplated—three new ones and six remodeled facilities.

Here is how the actual savings compared with the estimate:

Item	Estimate	Actual
Mileage Saving	$24,000	$29,300
Wage Saving	36,000	28,600
Gross Saving	60,000	57,900
Added Expense	15,000	15,500
Net Saving	$45,000	$42,400

Some of the differences between actual and estimated figures stemmed from such factors as unanticipated rise in wages and fuel costs and increased capital expenditures.

Decentralization is not yet complete since only nine garages have been constructed for the 17 localities, as shown in Fig. 2. Compare this with the old operation from two centralized garages, as shown in Fig. 1.

Other Gains Shown

The decentralized operation has shown other advantages in addition to the financial saving. These include the ability of the decentralized garage mechanic to give closer personal attention to the trucks under his charge as compared with receiving 60 to 75 trucks, all run into one garage at night requiring minor repairs. Often the least important of these are not made under the former setup.

Additionally, installing a water pressure pump at each decentralized garage provides for washing the trucks out in the open during nonfreezing weather. The washing is done by the combintion gasoline dispenser, greaser, and janitor. This together with stepped-up production in the company's paint shop helps improve appearance of the trucks. (Paper "Economics of Decentralized Versus Centralized Truck Operations," was presented at SAE Chicago Section, March 8, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Tells of Tire's Role In Earthworking Field

Excerpts from talk by

JOHN L. COLLYER

B. F. Goodrich Co.

DEVELOPMENT of rubber tires for farm tractors and implements represents a major change in the farming industry that has taken place within a span of less than 20 years.

Many may recall the early controversies. There was a genuine disbelief on the part of some that a rubber-tired wheel could ever do the job that a steel wheel could in a farming operation. But the change was made and there is no lingering echo of that early controversy.

The excellent teamwork between the tractor and rubber industries has established the fact that by sound design, certain different types of rubber tires can be created to do specific jobs. The early ideas which our industries had about the correct type and thickness of treads, for example, have been discarded. Each year has seen steady progress toward the type of tire which will represent to the farmer the best value in terms of low cost and efficient operation.

The SAE Tractor Technical Committee has been performing an outstanding service in the standardization and simplification of tires, and in the cooperation it has extended to the engineers of the rubber industry. It has a record of notable achievement, and I take this opportunity to congratulate the men who are responsible for so much progress in cost-reduction through sound cooperative planning and design.

The establishment of a practical minimum number of basic wheel sizes is highly desirable because of the economies that come through volume production of simplified inventories of tires, all of which can influence the costs and prices.

As to development of the off-the-highway tire, a few years ago it would have taken a vivid imagination to visualize what rubber tires are doing

daily in certain off-the-road services —carrying 80-ton loads out of coal pits or excavation pits, up steep grades and across all kinds of terrain. A single tire today can carry as much as 20 tons. Such a tire stands as high as 10 ft and may weigh more than a ton.

Generally speaking, tires made for earthmoving vehicles are of two types. One is the positive traction tire; the other is a tire for free-rolling wheels, where traction is not an important requirement. There are also several other types of tires which have been developed for specialized work in off-the-road service.

Few major classifications in the tire industry have shown such spectacular growth as that which includes these various types. The equipment on which they are used—and which, to a high degree, they made possible—came rapidly into favor as soon as its superiority for doing hard jobs economically was established.

You may be interested in the estimate that the off-the-road tire represents a sales potential in the United States of $50,000,000 a year. We look forward to the continuing development and expanded use of off-the-road tires, first, because vehicles with suitable pneumatic tires have definite advantages over other types of units in mobility and speed of operation, and second, because a continued high rate of construction is anticipated.

You may have noted the statement recently in the Engineering News-Record that at the end of June of this year, there was an almost unbelievable construction backlog of over $45,000,000,000 in the United States and Canada. Much of this construction—such as earth work, irrigation, drainage, streets and highways, railroad relocations, and airports—requires a tremendous amount of earth and rock moving.

However, the use of off-the-road tires is not restricted to what might be called "public works." Many industries are making use of pneumatic tire equipment for off-the-road service, and this is especially true of mining —whether underground, open pit, or strip mining—and logging and quarrying. (Talk "Progress in Rubber" was presented at SAE National Tractor Meeting, Milwaukee, Sept. 15, 1949.)

Goals in Truck, Bus Development

Based on paper by

MERRILL C. HORINE

Mack-International Motor Truck Corp.

MOST operators look for major operating economies in the vehicle itself, rather than in rerouting, rescheduling, and redispatching, which are factors which will raise productiveness of existing equipment.

Different combinations of axles, transmissions, powerplants, choice between gasoline, diesel or butane fuels, multispeed arrangements, hydraulic drives, superchargers, retarders, and novel suspensions are all avenues open to the truck and bus design engineer.

Undersize and underpower for large loads means shorter life, higher maintenance costs, and lack of safety for the vehicle, and even the fuel economy which is hoped for fails to materialize. It takes about so much fuel to produce a ton-mile.

There is no economy in cost saving alone. Greatest economy to the operator results from the greatest production of useful transportation in a given time at the lowest practicable unit cost. Total costs are significant only as they relate to total output of ton-miles.

Performance is simply the ability of a vehicle to overcome resistance to motion at a given speed. The greater the speed, the greater the resistance, and hence the greater power required. Principal resistances to be overcome are:

- Rolling,
- Grade,
- Air or wind, and
- Friction within the vehicle itself.

An SAE committee is studying these factors with a view to simplifying performance determinations. It is hoped that a uniform yardstick may be set up to compare results fairly.

One of the first objectives in forward-looking design is to provide sufficient power to get the performance potential, or the net horsepower to drive a given vehicle along a certain stretch of road at a given speed.

The fundamental requirement is power and not displacement, torque, or gears.

We are groping for an index to define the optimum relationship of power to gross weight. The figure of 270 lb per net hp has been suggested for over-the-road haulage.

Now 300 hp appears to be the practical limit for any size of highway vehicle, largely because of the restricted availability of practical engines with greater output.

If, however, there is economic justification of larger engines, such powerplants would be forthcoming. There is, so far, some doubt whether this much power is warranted. (Paper "Objectives in Truck and Bus Development" was presented at SAE Southern California Section, March 14, 1949. This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

High C. R. Engine Tougher To Make

Based on paper by

H. RICHARD O'HARA, JR.

Buick Motor Division, GMC

HIGH-SPEED engines capable of withstanding compression ratios of 10 to 1 impose many production problems. This type of engine inherently involves closer machining tolerances and more exacting manufacturing and inspection procedures.

Speculation has it that such engines will be shorter in overall length and height, of more sturdy construction, and will have a stroke-to-bore ratio of about one. These factors tend to facilitate machining. The main and connecting rod bearings will be larger to withstand the heavier loads. The main crankshaft and connecting rods probably will be forged to get the necessary rigidity characteristics.

The change-over to this high compression type of engine will require the following changes tightening up present production standards:

1. Barring any major advances in valve seat milling machinery, it probably will be necessary to lap all seats to the respective valves. Present machinery cannot mill or check valve seats to less than 0.004 in. total eccentricity with the valve stem guide.

2. Main and connecting rod bearings should be machined with a maximum taper and out-of-round of 0.0002 to 0.0003 in.—average present standard being 0.0003 to 0.0004 in.

3. Connecting rods should be held within ±0.005 lb on each end, providing a total tolerance of 0.020 lb on the net weight. This will require special machinery which is under development, but not presently used in production by any manufacturer. Best tolerance obtainable today is about ±0.0075 lb.

4. Pistons also will have to be machined to correspondingly low weight tolerances of ±0.004 lb. This tolerance should not impose the problem that the connecting rod limit does because we already have equipment which is

Cont. on p. 85

DIESEL

SAE National Diesel Engine Meeting

Nov. 1–2

The CHASE Hotel
St. Louis, Mo.

Tuesday, Nov. 1

9:00 a.m.

Welcome to St. Louis
A. J. JEUDE, General Chairman
L. A. WENDT, Chairman

Design and Development of the Lima-Hamilton Locomotive Diesel Engine
—F. J. GEITTMAN, Lima-Hamilton Corp.
Prepared Discussion

2:00 p.m.

A. J. JEUDE, Chairman

Operating Problems of Railway Diesel Locomotives
—R. W. SENIFF, The Baltimore and Ohio Railroad Co.

Rates of Wear in Railroad Diesel Electric Locomotives
—RAY McBRIAN and L. C. ATCHISON, Denver Rio Grande Western Railroad Co.
Prepared Discussion

4:15 p.m.

Inspection Trip to Terminal Railroad Association (See below)

Wednesday, Nov. 2

9:00 a.m.

RAYMOND TUCKER, Chairman

The Composition of Diesel Exhaust Gas
—M. A. ELLIOTT and R. F. DAVIS, Bureau of Mines

West Coast Smoke and Odor Problems
—L. J. GRUNDER, Richfield Oil Corp.
Prepared Discussion

1:00 p. m. (Following lunch)

Inspection Trip to St. Louis Public Service Co. (See below)

2:30 p.m.

ADAM EBINGER, Chairman

The Diesel Engine Cold Starting Problem
—F. L. NELSON and C. J. ULZHEIMER, Socony-Vacuum Oil Co., Inc.

Pioneering of a Diesel for Highway Cargo Transportation
—ADRIAN CURTIS, Garrett Freight Lines, Inc., and C. N. GUERASIMOFF, The Buda Co.
Prepared Discussion

Inspection Trips

Tuesday Nov. 1—4:15 p.m.-6:15 p.m.
Diesel engine shops and control tower, Terminal Railroad Association of St. Louis, Union Station.
Chartered buses leave Lindell Avenue entrance, Chase Hotel, for Union Station at 4:15 p.m., returning about 6:15 p.m.
Round-trip bus ticket $1—available at Registration Desk.

Wednesday, Nov. 2—1:00 p.m. (following lunch)—2:30 p.m.
Maintenance shops of St. Louis Public Service Co.—winners of ATA award for maintenance for the past several years.
Chartered buses leave Lindell Avenue entrance of Chase Hotel at 1:00 p.m. **sharp** for Park Avenue Garage; leaving Garage promptly at 2:15 to return in time for 2:30 p.m. technical session.
Round-trip bus ticket $1—available at Registration Desk.

DIN

Wednesday, Nov. 2

6:30 p.m.

Starlight Roof

F & L

SAE National Fuels & Lubricants Meeting

Nov. 3–4

The CHASE Hotel
St. Louis, Mo.

Thursday, Nov. 3

9:00 a.m.

Welcome to St. Louis

A. L. HEINTZE, General Chairman

R. O. SLATTERY, Chairman

The Lubrication of Railroad Diesel Engines
—E. F. H. PENNEKAMP, L. E. MOODY, and *H. L. BAKER, Standard Oil Development Co.
(*Now associated with Esso Standard Oil Co.)

Small Single Clyinder Engine Test for the Evaluation of Compounded Motor Oils
—M. R. CLAPP and J. L. PALMER, Lubrizol Corp.

Light Engine Oils for Improved Sub-Zero Operation (Report of the CLR Group on Engine Oils Lighter than SAE 10, Coordinating Research Council)
—V. G. RAVIOLO, Ford Motor Co., to present

2:00 p.m.

R. J. GREENSHIELDS, Chairman

Automatic Transmission Fluid, Type A, for Passenger Cars
—H. R. WOLF, Research Laboratories Div., General Motors Corp., and J. L. McCLOUD, Ford Motor Co.

NER

Joint Dinner Sponsored by National Diesel Engine and National F & L Meetings

W. S. RIGBY, Chairman
W. J. DAVIDSON, Toastmaster
S. W. SPARROW, SAE President

"Future Problems of the Internal Combustion Engine"
C. F. KETTERING
Director, General Motors Corp.

Friday, Nov. 4

9:00 a.m.

L. D. BOYCE, Chairman

Evaluating Lubricating Oil Through Fleet Testing
—W. S. HOOCK, Sinclair Refining Co.

Oil Ring Plugging—A Missing Link in Laboratory Tests
—L. J. TEST, Atlantic Refining Co., and C. A. HALL, Ethyl Corp.

Engine Sludge and Varnish (Report of the Engine Varnish & Sludge Group, Motor Fuels Division, CFR Committee, Coordinating Research Council)
—H. J. GIBSON, Ethyl Corp., to present

2:00 p.m.

A. L. HEINTZE, Chairman

Symposium on Gear Lubricants

Performance Testing of Gear Lubricants
—W. J. BACKOFF, N. D. WILLIAMS, and K. BOLDT, Pure Oil Co.

Moisture Corrosion Test for 2-105B Gear Lubricants
—T. P. SANDS, Monsanto Chemical Co.

Laboratory Wear Tests With Automotive Gear Lubricants
—S. A. McKEE, J. F. SWINDELLS, H. S. WHITE, and W. MOUNTJOY, National Bureau of Standards

About

KOEPNICK

RUMMEL

R. W. RUMMEL has been appointed chief engineer for Trans World Airline. He was formerly manager of aircraft procurement, and succeeds **L. R. KOEPNICK**, who resigned as chief engineer to accept a position in the Army Air Force. Koepnick will continue to serve as Meetings chairman of the Air Transport Activity. Rummel joined TWA in 1943, and has since served in various engineering capacities.

T. A. WELLS, vice-president in charge of engineering, Beech Aircraft Corp., is again world's champion skipper of Snipe-class sailing, having won the International Championship in competition with the champions of eight other countries at the Larchmont (N. Y.) Yacht Club late in August. Wells first won the International title in 1947 when the contest was held at Geneva, Switzerland. . . . Earlier in August at Jamestown, N. Y., Wells had won the U. S. National title, which qualified him for the International competition. . . . In the U. S. Nationals, Jean Blanchard, daughter of **DON BLANCHARD**, manager, technical committee division, SAE headquarters staff, was eight in the field of 48 craft.

WILLARD H. JENSEN is now a tool designer for Texas Engineering & Mfg. Co., Inc., Grand Prairie, Tex. He had been a tool designer with R. G. LeTourneau, Inc., Longview, Tex.

P. J. MILLER has been elected a vice-president of the Toledo Heater Co., Toledo. He had been general manager of the company's Bus Equipment Division for the past year, and previously was assistant works manager of Evans Products Co., Detroit.

EARL W. HAEFNER who has been manager of the parts and service department of the Airtemp Division, Chrysler Corp., Dayton, Ohio, has joined Borg-Warner International Corp., Detroit, as service manager.

DONALD LEE GRISHAM, who had been attending the University of Colorado, has joined North American Aviation, Inc., at Los Angeles as engineering draftsman.

NORMAN F. LANG is now a junior engineer with Curtiss Propeller Division, Curtiss Wright Corp., Caldwell, N. J.

J. L. S. SNEAD, JR. has been appointed evecutive vice-president of Consolidated Freightways, Inc. He is vice-president of the SAE Transportation and Maintenance Activity Committee and a member of several other SAE committees.

EDGAR F. WAIT has been named assistant plant manager of Ford Motor Co.'s Highland Park operations. He joined Ford in 1936 to organize and build the Ford Tire plant and remained in charge of tire manufacturing until 1942, when the government purchased the plant. He then became a Ford Motor Co. general superintendent, and in 1945 was appointed director of manufacturing services.

PAUL HUBER has been named research engineer of the Fram Corp., Providence, R. I. Huber, from 1946 until his recent affiliation with Fram, was director of research and experimental activities of Willys-Overland, and formerly chief engineer of the General Motors Proving Ground at Milford, Mich.

S. H. FEDAN, manager of the Aeromatic Propeller Department of Koppers Co., Inc., since 1941, has been named assistant to the sales manager of the company's Metal Products Division. He will continue to direct the activities of the Aeromatic Propeller Department.

GEORGE T. LADD has been appointed production engineer of the Al-Fin Division of Fairchild Engine & Airplane Corp., Farmingdale, N. Y. He will work on the design and development of bonded bi-metallic products now being offered by Al-Fin and its six licensees. Ladd had been affiliated with Wright Aeronautical Corp., Wood-Ridge, N. J., for the past 11 years, as a project engineer.

DAVID W. RICE is now sales engineer in the Instrument Division of Thomas A. Edison, Inc., West Orange, N. J. Prior to this post he was assistant project engineer at Wright Aeronautical Corp., Wood-Ridge, N. J.

J. W. PRIMEAU has been elected president of the newly established B-W-H Service Parts, Ltd., Merritton, Ont., Canada. This company has been formed to distribute a complete range of Borg-Warner automotive replacement parts throughout Canada. Primeau is also vice-president and general manager of Hays Steel Products, Ltd., of Merritton.

CHESTER E. MINES is now associated with the Allison Division of General Motors Corp., Indianapolis, Ind.

ROBERT E. ALLEN, vice-president of Plastic & Rubber Products, Inc., who has had charge of the firm's Chicago office for the past four years is now transferring his activities to the home office in Dayton, Ohio. His transfer has been made in connection with the firm's $125,000 expansion program.

ROBERT A. WOODWARD is now estimating engineer for the Columbus Bolt & Forging Co. in Columbus, Ohio.

Members

KENNETH W. CARLSON, a recent graduate from the University of California, Berkeley, has joined Boeing Airplane Co., Seattle, Wash., as a junior engineer on design.

GEORGE E. ROWBOTHAM, who had been standards engineer for American Coach & Body Co., Cleveland, has been appointed standards engineer by Ford Motor Co., Dearborn. He is in the engineering laboratory.

RALPH N. DuBOIS, who has been associated with the Instituto Technologico de Aeronautica at Rio de Janeiro, Brazil, for several years, is planning a visit to the United States next January when he expects to attend the 1950 SAE Annual Meeting in Detroit. DuBois returned to Rio recently from a month's vacation in Argentina, where he spent most of that time in the Andean Lake district. He is a past-chairman of the Detroit Section.

GEORGE V. BIANCHINI, who has been attending Michigan College of Mining and Technology, Houghton, has joined the testing engineer staff of Allison Division, General Motors Corp., Indianapolis.

HAROLD A. MINGES has resigned as manager of the Service Manual Department of Baldwin Locomotive Works to become design engineer with Witte Engine Works, diesel engine manufacturers, Kansas City, Mo. The company is a division of Oil Well Supply Co., a U. S. Steel Corp. subsidiary.

JAMES A. GRAVES, who has been attending the University of Oklahoma, has joined the Engineering Department of Stanolind Pipe Line Co., Tulsa, Okla.

RICHARD CRETER, chairman of Metropolitan Section, has joined U. S. Tire Supply Co., New York City, as general service manager, according to an announcement by **KENNETH R. SCHAAL**, president. Creter had been general service manager of Cummins Diesel Railroad Equipment Co., Inc.

JOHN D. TEBBEN, Detroit sales consultant, has been retained by Weldaloy Products Co., of that city as consultant. He is a former vice-president of the American Welding Society and past-chairman of AWS Detroit and Indiana Sections.

MARCEL F. DeMULLER, president of the Willys-Overland Export Corp., has been decorated with the order of Nichan-Iftikhar by the Bey of Tunisia for his efforts to improve world trade conditions. The order was established 118 years ago and is Tunisia's highest accolade.

THEODORE R. JAMIESON, II has been appointed automotive body designer of Oldsmobile Division, General Motors Corp., Lansing, Mich. He had been a body designer with the corporation's Cadillac Division in Detroit.

SEMON E. KNUDSEN has been appointed director of the Process Development Section of the facilities and processes staff, General Motors Corp. He has held executive production posts at GM's Pontiac and Cadillac Divisions. He is the son of the late Gen. William S. Knudsen, one-time president of GM.

HARRY B. SEED has been named sales and service representative of Wright Aeronautical Corp. in Los Angeles. He had been assistant service manager of Chandler Motor Car Co., Cleveland.

CHARLES A. AMANN, formerly an instructor at the University of Minnesota has joined the Research Laboratories Division, General Motors Corp., as a graduate in training.

ROBERT A. WALKER has joined AiResearch Mfg. Co., Los Angeles. He had been manager of aircraft engineering for Transcontinental & Western Air., Inc. Kansas City, where he had served successively as vice-chairman for aeronautics, vice-chairman, and last year chairman of the SAE Kansas City Section.

GORDON F. DANHOF, who graduated last June from Michigan State College, has joined Minneapolis Honeywell Co., Minneapolis, as sales engineer.

ROBERT G. LUSIAN has been named aerodynamacist engineer with Consolidated Vultee Aircraft Corp., Fort Worth, Tex. He had done similar work when with Curtiss Wright Airplane Division, Curtiss Wright Corp., Columbus, Ohio.

THOMAS E. PAXTON has been named an inspector of refinery units for Gulf Oil Corp., Philadelphia.

JAMES D. MOONEY has been appointed chairman of the board for Technical Managers, Inc., industrial consultants in New York City. He recently resigned as chairman and president of Willys-Overland Motors, Inc. **JAMES W. SCHIEFER** has been appointed vice-president of Technical Managers. He was previously production manager at Willys-Overland.

MOONEY

SCHIEFER

GEORGE H. KUBLIN, formerly chief engineer of General Motors Overseas Operations, has been appointed manager of the Detroit staff of the Overseas Operations. **ALBERT A. MAYNARD** has been appointed chief engineer to succeed Kublin. Maynard was formerly engineer in charge of a Special Car Project for General Motors Product Studies.

KUBLIN

MAYNARD

BERSIE

KEATING

HUGH G. BERSIE has been appointed director of products and **JOHN K. KEATING** has been appointed assistant transportation product manager for the American Seating Co., Grand Rapids, Mich.

W. M. HOLADAY, Socony-Vacuum Oil Co., Inc., will talk on "Fuels—Their Present and Future Utilization" at the annual meeting of the American Petroleum Institute on Thursday morning, Nov. 10 in Chicago. In relation to efficiency and resources' conservation, he will discuss the future uses of coal, liquid fuels and gas, as well as the question of what should be the fuel for internal combustion engines—diesel fuel or gasoline. . . . Along with a paper by Walter G. Whitman, Massachusetts Institute of Technology, on "Liquid Fuel Supplies and National Security", Holaday's paper will open a forum held under the joint auspices of the Production Refining and Marketing Division of API. Whitman will discuss responsibility for national security, what fuel requirements of the next war may be, and how the characteristics of probable future fuels will affect military fuel production. . . . **K. G. MacKENZIE**, Texas Co. will be chairman of the session at which the Holaday and Whitman papers are presented.

JEROME S. BUZZARD is now mechanical design engineer with the Western Electric Co., Winston Salem, N. C. He had been a layout draftsman wtih Packard Motor Car Co., Toledo. His new work is in the radar field.

HOWARD W. CRUSEY, formerly field engineer with Thompson Products, Inc., Cleveland, has joined the A1-Fin Division of Fairchild Engine & Airplane Co., Farmingdale, Long Island, N. Y., as sales engineer. He had served as a plant representative of the SAE Cleveland Section.

HARRY O. HILL has been appointed general manager of Jordan Mfg. Co., Lake Odessa, Mich. For several years he was a member of the SAE Diesel Engine Activity Committee. He had been sales manager of Diesel Engine & Mfg. Co., Chicago.

BRUCE W. CARKIN, a recent graduate from Oregon State College, has been named an experimental test mechanic for Electro-Motive Division, General Motors Corp., LaGrange, Ill.

EDWARD M. BONETTE has joined Cummins & Moran, diesel engine distributor at Phoenix, Ariz. He had been assistant regional manager of the Cummins Engine Co. in Los Angeles, where he served as vice-chairman of Diesel Engineering of the SAE Southern California Section and Field Editor of the SAE Journal.

DONALD G. MOORE has been appointed supervisor of vocational education at the Folsom State Prison at Represa, Calif. He had been instructor of automotive mechanics at the Hanford (Calif.) High School, and has written portions of apprentice texts used in that state.

WARREN E. BURNHAM, who graduated last June from the University of Wisconsin, has joined the New York Department of Public Works, Albany, as a junior heating and ventilating engineer.

CAROL SZABADOS, formerly chief engineer of Leonida & Co., SA, has been appointed chief engineer of the Roumanian State Railway at Bucharest.

A. O. PAYNE has been appointed chief engineer of the Oil Equipment Division of A. Y. McDonald Mfg. Co., Dubuque, Iowa. He had been chief engineer of the OPW Corp., Cincinnati, and in the early days of the SAE St. Louis Section had served as chairman.

THOMAS L. FAWICK, president of Airflex Clutch Co., Inc., Cleveland, has won this year's John Price Wetherill Medals of the Franklin Institute. Established in 1925 the medal is awarded for his inventions and discoveries in physical sciences or for new and important combinations of known principles. The clutch he invented will be cited at the presentation on Oct. 19 in Philadelphia.

A. J. MAURICE OUSTALET, JR., has been appointed sales manager of the Truck & Equipment Department of Capital City Ford Co., Inc., Baton Rouge, La.

BENJAMIN G. SYMON, president of the National Lubricating Grease Institute, Kansas City, Mo., announced that the organization's 17th Annual Meeting will be held Oct. 3 to 5 in New Orleans. Among the authors is **NORMAN L. KLEIN**, Office of the Chief of Ordnance, Department of the Army.

FRANK W. GODSEY, JR., has been appointed manager of the new Special Products Development Division of Westinghouse Electric Corp., Pittsburgh. He is co-author of "Gas Turbines for Aircraft" recently published by McGraw-Hill Book Co.

HOWARD G. HEINIG has been named assistant chief engineer of Alma Trailer Co., Alma, Mich. He is a recent graduate of Tri-State College, Angola, Ind.

SYLVANUS C. WALKER is now production supervisor with Cherokee Mfg. Co., Cherokee, Okla., manufacturers of farm machinery. He graduated last June from Oklahoma A & M College.

MILTON KAMINS, a recent graduate from M.I.T., has joined the Studebaker Corp., South Bend, Ind., as an experimental engineer.

JOHN C. RHODES is tool designer and draftsman with Aluminum Industries, Inc., Cincinnati. He is a graduate of the University of Cincinnati.

C. WILLIAM LAUBIN has joined Fredric Flader, Inc., North Tonawanda, N. Y. as a consulting design engineer. He had been assistant technical director of the Phillips Laboratories, Dobbs Ferry, N. Y.

NIKOLA TRBOJEVICH has joined the faculty of Lawrence Institute of Technology, Detroit. He had been research and consulting engineer with the National Broach & Machine Co. of that city.

JOHN F. ELSENHEIMER has joined the Wright-Patterson Air Force Base in Dayton as an engineer.

JAMES E. COMER, who graduated last June from Northrop Aeronautical Institute, has joined the American National Insurance Co., Dallas, Tex.

CHARLES M. MARLATT has joined the engineering staff of the Barr Rubber Products Co., Sandusky, Ohio, as development engineer. He had occupied a similar position with the Cooper Tire & Rubber Co. at Findlay.

NORMAN E. DANIELSON has joined McDonnell Aircraft Corp., St. Louis, as final assembly inspector.

THOMAS R. BLAKESLEE has been transferred to the eastern area as sales representative and sales engineer for Federal-Mogul Corp. He had been in the Chicago area for the Detroit manufacturing company.

WALTER CRAFTS wrote the recently-published book "Hardenability and Steel Selection," together with John L. Lamont. Published by the Pitman Publishing Corp., coordinates hardenability theories and calculations. The hardenability calculations are presented for predicting the behavior of the material in the four steps of the process: (1) quenching rates, (2) depth of hardening in quenching, (3) softening and toughening in temperature, and (4) effects of chemical composition on mechanical properties. Price of the book is $5.50.

CHARLES H. CHATFIELD and C. FAYETTE TAYLOR, together with Shatswell Ober, have revised their textbook "The Airplane and its Engine," and McGraw-Hill Book Co., Inc. has just brought it out in its fifth edition, at $4.50. The book discusses fundamental principles, construction, and capabilities of the airplane and its engine. Treatment is descriptive, rather than mathematical. Technical terms abound, but each is introduced with a simply worded definition. An extensive bibliography supplements the text.

CHARLES R. FOX has joined the Aluminum Co. of Canada, Ltd., as an engineer at the Arvida, Que., plant. He graduated last May from the University of British Columbia, Vancouver.

WILLIAM J. MILLER has been appointed equipment engineer with the Washington State Highway Department at Olympia. He had been sales engineer with Murphy White Trucks, Seattle, Wash.

EMERSON D. OGLE has been appointed manager of the Industrial Sales Department of SKF industries, Inc., Philadelphia. He joined the company in 1937 upon graduation from the University of Maryland.

IRVING I. FELDMAN, owner of the Aero-Automobile Engineering Co., Detroit, announced combining his company with Acs Electrological Laboratory, Buffalo, which has developed a new method which combines multiple inspection, weighing, measuring, and packaging to micro limits at high speeds.

NORMAN J. LAW has been named manager and vice-president of United Automotive Service, Harrisburg, Penna. He had been service and sales representative of American Bosch Corp., Springfield, Mass.

LADISLAW G. SROGI, formerly senior engineer with Packard Motor Car Co., Toledo, has joined Bell Aircraft Corp., Niagara Falls, N. Y., as a design engineer.

FREDERICK C. REED has joined Consolidated Vultee Aircraft Corp., Fort Worth, Tex., as a design engineer on hydraulics and landing gear. He had been doing similar engineering for the Airplane Division, Curtiss-Wright Corp., Columbus, Ohio.

CARROLL D. FRUTH has been appointed flight test engineer of Allison Division, General Motors Corp., Indianapolis, leaving a similar position with the Airplane Division of Curtiss-Wright Corp., Columbus, Ohio.

RODNEY L. BOYER, formerly project engineer with M. W. Kellogg Co., Jersey City, N. J., has joined the Applied Physics Laboratory of Johns Hopkins University, Silver Spring, Md., as administrative assistant.

STANTON R. COOK, a recent graduate from Northwestern University, Evanston, Ill., has joined Shell Oil Co., Inc., as a junior salesman in the Chicago area.

Continued on page 85

OBITUARIES

WALTER BERGIUS

Walter Bergius, who founded his own automobile manufacturing company in Glasgow in 1904 when he was only 23, died June 4. He still headed the company which has produced thousands of Kelvin diesel engines for fishing boats and yachts.

Since 1923 the company, Bergius & Co., Ltd., has enjoyed world wide distribution of its diesel powerplants which he designed.

His interest in yachting led him to design a series of standard motor boats, nearly 1200 of which were built before the beginning of World War II in 1939. He also designed his own motor yachts, Vailima I and II.

Bergius was enthusiastic about American machine tools and manufacturing methods, and was credited with a number of valuable tooling design ideas.

He worked closely with Sir Harry Ricardo on his improved engine designs. One line of the Kelvin engines was the Kelvin-Ricardo.

JOHN J. ELDRIDGE

John J. Eldridge, active for many years in the work of the SAE Cincinnati Section, died suddenly on May 29 in his home in Westwood.

He had been ardent in civic, church and fraternal work in Cincinnati since 1917 and had organized and headed the Buckeye Service & Supply Co., a leading engine rebuilder. He had been president of the Engine Rebuilders Association and later served as secretary of that national organization.

Eldridge entered the automotive industry as an apprentice machinist, and became service manager for Robert N. Cutting Co., Chicago, White truck distributor. His experience led him to the fleet superintendency of the Schultz Baking Co. where he was in charge of purchasing automotive equipment, designing and building bodies, and was in charge of maintenance, construction and assembly of the baking plant machinery and equipment.

During World War II the Buckeye concern devoted most of its time grinding hydraulic press crank shafts for Government contractors. His company had branches in Covington, Ky., and Lawrenceburg, Ind.

JOSEPH VAN BLERCK, SR.

Joseph Van Blerck, Sr., one of the nation's pioneer marine engine manufacturers, passed away Sept. 5 at Fort Lauderdale, Fla. He was 74.

Before embarking on his marine engine designing and manufacturing career, he was associated with Henry Ford and worked on the development of early Ford cars. After the day's work, Ford and his young son Edsel spent many pleasant evenings when he was building his first marine engine in the backyard of his Detroit home.

During World War I he built hundreds of engines for the U. S. Navy, and was selected by Secretary Josephus Daniels to design a special standardized engine for the Navy's lifeboats and launches.

Many of these, and later models of diesel marine and stationary powerplants are being used in this country and abroad.

He joined the Society in 1917, and since moving his business to Long Island had been a frequent and provocative discusser of diesel engine topics at Metropolitan Section meetings.

SAE Section News

Baltimore Section Sees USN Experiment Station

• Baltimore Section
R. L. Ashley, Field Editor

September 8—Lt. (jg) K. Y. Saunders transported 85 members and their guests aboard the USS LCI 1032 from Ft. McHenry, Baltimore to the U. S. Naval Engineering Experiment Station, Annapolis, Md. and back. During the two and a half hour cruise groups were conducted through the ship, after which all were served delicious Navy chow.

At the Naval station Commander Hirschberger appointed five escorts to guide groups through the six major laboratories. These were the chemical, internal combustion engine, mechanical, metallurgical, wave mechanics, and welding laboratories.

Spokesmen explained the various operations of their respective labs. For example, in the wave mechanics lab one group saw a "shock table," built similar to deck plating, suspended at floor level. Many types of equipment used on ships are secured to this plate to test the equipment reaction to underwater explosive shocks. A 5-ft drop hammer, propelled by gravity, swings beneath the floor and strikes the bottom of the table. Any physical damage to the equipment can readily be observed and corrective measures taken. The benefits of such testing are boundless.

Commander Hirschberger explained that this, the Navy's oldest laboratory of such a type, is designed for applied research and testing rather than pure research. He also stated that the station is looking for young college graduates who wish an interesting and stimulating career.

St. Louis Section Tours McDonnell Aircraft Corp.

• St. Louis Section
G. C. Husbands, Field Editor

Sept. 13—St. Louis Section made an inspection trip, accompanied by SAE enrolled students from Parks College, of the McDonnell Aircraft Corp.'s facilities.

The trip terminated at the ramp, where the visitors watched an exhibition flight of "Little Henry," an Air Force ramjet aerial motorcycle.

Later, they viewed movies of recent McDonnell developments, including the Phantom and Banshee jet fighters, a twin-engine helicopter, and the XF-85 parasite jet fighter.

British Car Features Contrast with American

• Hawaii Section
Rene Guillou, Field Editor

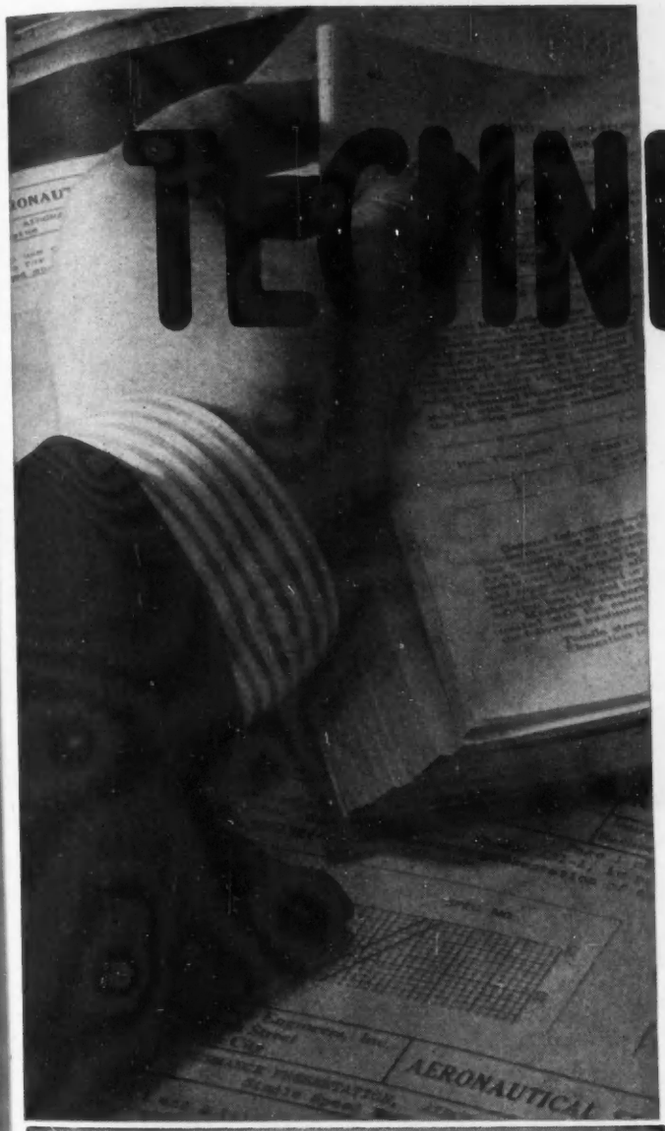
Aug. 15—British cars, particularly small British cars, are being sold to Americans who are dissatisfied with certain features of current American cars, **Robert McGill** pointed out in his talk on "The British Car in America."

McGill is automotive advisor for the Hawaiian Ordnance Depot.

Aside from the small size of the British cars, their most distinctive feature is a difference in steering "feel," due to less weight on the front wheels, smaller tires, and much lower steering gear ratios, according to McGill. It was also pointed out that the British use of an engine designed for performance at high rotative speed, in connection with a manual four-speed gear shift, places excellent car performance at the disposal of the driver who prefers doing things himself to having them done automatically.

Although the American car is outstanding in its ability to make long, high-speed runs over modern American highways with a minimum of driver fatigue, Hawaii, with its short distances and comparatively low speeds, has encouraged drivers who prefer British car characteristics, he said.

USS LCI 1032 puts in at Annapolis on Sept. 8, with Baltimore Section members aboard. They spent the day visiting the US Naval Engineering Experiment Station there



TECHNICAL COMMITTEE PROGRESS

Modify SAE Standard On Involute Serrations

THE SAE STANDARD on Involute Serrations recently has been revised and expanded to make it more useful and more consistent. Prepared for inclusion in the 1950 SAE Handbook by a subcommittee of the SAE Parts and Fittings Committee, the revised Standard will form the basis for a new American Standard on Involute Serrations.

Scope of the Standard has been increased to include finer pitches to supply the smaller sizes needed in the aeronautical industry. Sizes have been increased for diametral pitches of 40/80, 48/96, 64/128, and 80/160, and the range of the 32/64 pitch has been moved downward to agree with the coarser pitches.

Effective fits and allowable errors also have been introduced in the Involute Serrations Standard on the same basis as in the SAE Involute Spline Standard, now being revised. To include effective fits, the pin measurements are now given as maximum for the internal and minimum for the external. To calculate all fits, it is now necessary to have basic measurements between and over pins. New tables for this are included as well as additional tables giving maximum tooth space and minimum tooth thickness.

Nomenclature has been made to agree with both the SAE Involute Spline Standard and the practice of the American Gear Manufacturers Association.

This revision typifies the evolution of a standard of this type, notes Subcommittee Chairman C. H. Stanard. It consists of refinements and modifications based on practical application and experience

SAE TECHNICAL BOARD

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with the Standard, both from product and tooling viewpoints.

The Involute Serrations Standard first was adopted by SAE in 1947. The old SAE Serrated Shaft Standard that preceded it had several inconsistencies. For example, many serrations, especially those generated by hobbing were found to have an involute form, even though they were referred to as straight. This was one of the factors that led to the replacement of the old SAE Standard with one covering a 45-deg involute serration.

Size Range Increased

Even diametral pitches were adopted in the SAE Involute Serrations Standard, approved in 1947, which are silghtly coarser than those in the old SAE Serrated Shaft Standard. In the range of tooth sizes, diameters both larger and smaller than those in the old Standard are given, which offers a much greater selection of sizes.

Serving with Chairman Stanard on the subcommittee of the SAE Parts and Fittings Committee, which prepared the current revision, are: G. L. McCain, Chrysler Corp., subcommittee vice-chairman; H. H. Gotberg, Colonial Broach Co.; J. P. Breuer, Barber-Colman Co.; Harry Pelphrey, Michigan Tool Co.; W. A. Siler, Delco-Remy Division, GMC; J. F. Cramer, Boeing Aircraft Co.; and Glenn Tinkle, Illinois Tool Works.

Aero Accessory Group Seeks Vibration Data

DATA on vibration characteristics of aircraft engines are being sought by the SAE Aircraft Accessory Flanges and Drives Committee E-24.

Committee members consider such information necessary for the intelligent design of accessories which are subjected to engine torsional and linear vibration in service. While such information is scattered throughout the industry, the Committee believes that no effort has been made to collect, classify, or interpret it.

For this reason the SAE Committee plans to circulate a questionnaire among engine manufacturers to gather presently available data on torsional and linear vibration characteristics of engine drives for reciprocating, turboprop, and turbo-jet powerplants.

Committee members visualize, as the end product of this study, an SAE aeronautical information report to serve as a general guide to accessory and aircraft manufacturers. They plan to develop data on general engine types rather than on specific engines.

For example, this is how the Committee hopes to issue the information it collects: "Double-bank radial engines, nine cylinders per bank, usually have between 0.005 and 0.010 in. rear case vibration, peak-to-peak, of the 3½ order, at cruising speed. This 3½ order produces the largest excitation and it must be avoided in designing accessories for critical frequency."

The Committee, headed by N. F. Rooke, Pratt & Whitney Aircraft, is also developing an aeronautical recommended practice covering various quick attach-detach devices for mounting aircraft engine accessories. Timing on two airplane installations revealed that such devices take less than one minute to attach or detach, excluding electrical connections.

Other projects under way are ARP's for power transmissions, for driving remotely mounted accessories, and a definition of the clearance required for generator shrouds to be incorporated in the generator installation clearance drawings.

You'll Be Interested to Know . . . .

REPRESENTATIVES OF MAJOR ENGINEERING SOCIETIES, under Engineers Joint Council auspices, plan to explore "the desirability and practicality of increased unity in the engineering profession, and possible forms of organization which may be considered." Past-President W. S. James, vice-president, Fram Corp., will represent the SAE in the discussion group.

SAE HAS NAMED R. R. FALLER, Ethyl Corp., as its representative to consult with the Inter-Society Committee on Engineer Student Chapters. The Committee, formed at the invitation of the American Society of Civil Engineers, plans to explore proper means "of enabling student chapters to function without confusion and duplication."

THE SAE RECENTLY BECAME A SUSTAINING MEMBER of the U. S. National Committee, of the International Commission on Illumination. This grows out of SAE participation in the Committee's work to achieve greater international uniformity on motor vehicle lighting.

MANY SAE MEMBERS WILL BE ACTIVE in the 31st National Metal Exposition and Congress at the Cleveland Public Auditorium, Oct. 17-21. Walter E. Jominy, Chrysler Corp., is vice-president of the American Society for Metals, and G. N. Sieger, S. M. S. Corp., is president of the American Welding Society—two of the organizations sponsoring the Congress. W. R. Breeler, Allegheny-Ludlum Steel Corp., and Elbert S. Rowland, Timken Roller Bearing Co., will be among those acting as joint chairman at two of the technical sessions, while papers will be read by James D. Armour, Union Drawn Steel Division, Republic Steel Co.; Merrill A. Scheil, A. O. Smith Corp.; John M. Lessells, M. I. T.; and L. A. Danse, General Motors Corp. . . . E. H. Dix, Jr., Aluminum Co. of America, will deliver the Edward deMille Campbell Memorial lecture scheduled during the Annual Meeting of the American Society for Metals which is a part of the Congress.

THE ROAD RATING EXCHANGE PANEL DATA for 1948 indicated that about 70% of the road ratings of motor fuels made by individual company laboratories were within plus or minus two octane numbers of the average, the conference of the Detonation Road Test Group of the CFR Motor Fuels Division of the Coordinating Research Council found during its July 11 to 22 session in Detroit. . . . Primary purpose of that conference was to study ways to reduce deviations being obtained—and effects of personnel, instrumentation, and techniques particularly. Chairman of the Road Test Group, made up of 79 representatives of refiners and vehicle manufacturers, was Thomas H. Risk, Ethyl Corp. Two Londoners were present.

About SAE Members

Cont. from p. 81

L. L. HENRY, assistant dean of the college of engineering, Detroit Institute of Technology spent part of his summer vacation as a student at the Diesel Engine Symposium at the University of Wisconsin, Madison. The program included a number of plant inspections and was organized with the cooperation of the Diesel Engine Manufacturers Association.

GEORGE CODRINGTON, a vice-president of General Motors Corp. and general manager of the Cleveland Diesel Division, Cleveland, Ohio, and **KNUT O. KEEL,** chief engineer, returned from a month's business trip in Europe. They observed trial runs of the Mui Ann, a 3500 ton passenger-cargo vessel built at Kristiansands, Norway, and powered with four Cleveland diesels producing 3000 shp. This is the first diesel-electric built in that country. The ship will ply between China and Siam.

J. A. DURR, recently resigned as technical advisor to the general manager of the Albion Malleable Iron Co. of Albion, Mich., has joined the staff of the Erie Malleable Iron Co. of Erie, Pa.

H. W. OVERMAN has joined the American Pulley Co., Philadelphia, Pa., as sales manager of the Materials Handling Division.

HEINZ HANAU is now development engineer with Walter Kidde & Co. in Belleville, N. J. He was previously a design engineer with the Bell Aircraft Corp. in Buffalo, N. Y.

PHIL C. FAITH has joined Transmission & Gear Co., Inc., Dearborn, Mich. on production control work. He had been chief equipment inspector of Fifth Avenue Coach Co. & Affiliates, New York City.

ROBERT A. PAINTER has been appointed senior engineer with Air Associates, Inc., Teterboro, N. J. He had been stress engineer of Spicer Mfg. Division, Dana Corp., Toledo.

STUART K. WOOD is now a junior engineer with Boeing Airplane Co., Seattle, Wash.

KENNETH G. MAGISTRATE has joined Stanley F. Chamberlain Co., New York City as a technical writer. He graduated last May from the Aeronautical University, Chicago.

HERMAN L. EBERTS is now construction engineer with Canadair, Ltd., aircraft manufacturers in Montreal, Que., Can. He had been director of sales and engineering for the Federal Electric Mfg. Co. of that city.

HARRY J. DOYLE has been appointed small arms technical advisor with the U. S. Army Group in Greece. He had been general parts & service supervisor of Starr Dilger Motor Sales, Toledo. He will also serve as technical director of automotive schools he will set up under the four-year ECA program in that country.

HENRY A. SHERWOOD has been appointed vice-president in charge of engineering of Uni-Flow Engine Corp., New York City. He had been president of Industro-Matic Corp. of America, also of New York.

ALFRED M. MORROW has joined the New York Board of Education's Bureau of Plant Operation and Maintenance, Brooklyn, as mechanical engineering draftsman.

High C.R. Engine

Cont. from p. 75

able to hold this limit accurately on a production basis.

5. Weight of piston and connecting rod assembly will be held to the lowest amount commensurate with required strength, which means that the pistons probably will be made of aluminum alloy. Costs of machining aluminum compare with those for machining cast-iron pistons, so the major price differential between the two types lies mainly in the higher cost of aluminum alloy.

6. The engine assembly must be balanced within one-sixteenth of an ounce-inch to be satisfactory, which also creates production problems and requires development of new machinery. Present equipment is capable of balancing within one-eighth of an ounce-inch.

Such close weight limits on reciprocating parts are necessary to the undesirable secondary unbalance condition at the higher speeds at which the future engine will operate. This unbalance cannot be corrected after the engine is assembled.

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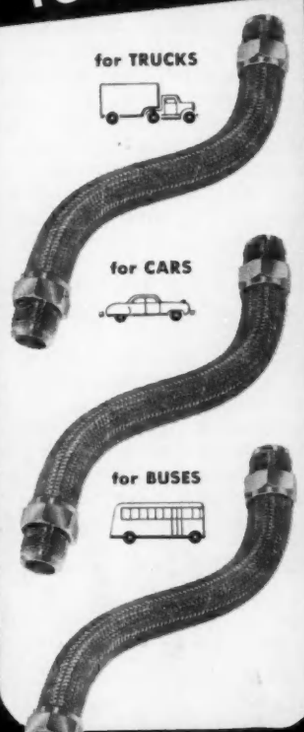

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is the design of combustion chambers in the cylinder head to be either cast or machined. A chamber cannot be cast within volume limits of less than ±0.1 cu in. on a production basis, due to inherent variations in casting which result from factors like warpage, sands, cupola mixes, and permeability of molds. Very likely a variation of 0.2 cu in. could be tolerated at compression ratios less than 8 to 1; it would be more economical to use cast chambers in this range.

As ratios are increased, it will be necessary to hold variations to progressively smaller increments to maintain the same percentage of volume ratio and to get a desirable balance condition. At such time the machined chamber will be required. It also will be more economical than trying to hold a cast chamber by grit blasting or other methods in production, with ensuing scrap.

In most cases, changes in manufacturing methods will result in an abrupt departure from past procedures and will entail considerable equipment and tooling expenditures. If the present engine manufactured by the company is adaptable to high compression —such as a valve-in-head V-8 or straight six—some of the existing equipment might be usable to manufacture the new engine. But in most cases the equipment will be obsolete.

This stems directly from development work on continuous or process type machinery. The advantages provided by this type of machining are exemplified mainly by reductions in labor costs and elimination of scrap due to inherent human errors. But when new equipment has to be acquired on a large scale, or where expense of reworking the old equipment is prohibitive, this new type of machinery satisfies most requirements.

Lacks Flexibility

Major disadvantage of this machinery rests in the fact that it lends itself to specialization and is not easily adaptable to design alterations once installed. This factor should be kept in mind when procuring equipment to avoid obsolescence as much as feasible.

Due to the high cost of such an extensive retooling program and considering the petroleum industry's problems in increasing octane ratings of fuels, I believe we can expect any manufacturer who has a valve-in-head engine at present to retain that engine, with slight modifications, for at least another one or two-year period.

This time might also serve to realize a readjustment in material and equipment costs, which would act in the automobile manufacturer's favor. (Paper "Manufacturing Problems Created by High Compression Engines," was presented at SAE Detroit Section, May 2, 1949. This paper is available

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Application Defines Diesel Air Cleaner

Based on paper by

FARR CO.

SELECTION of diesel engine air cleaners hinges on factors such as engine air needs, dust encountered, and installation requirements.

Air cleaner size depends largely on the volume and velocity of the air which must pass through the cleaner. It is simple to determine the amount of air the engine will require, but tough to convert this value into the air flow requirements of the filter.

Air speed through the cleaner affects considerably the cleaning efficiency. Therefore, it is necessary to establish the peak or surge velocity as established by the engine design.

For example, a 1-cyl four-cycle engine may require an air volume of 200 cfm as computed on the basis of displacement. Yet because air is taken into the combustion chamber on every fourth stroke only, air cleaner must be computed on the basis of the time the air actually is flowing into the cylinder. This peak velocity will vary with the number of cylinders, type of engine, rpm, and other factors. In many cases, particularly on multi-cylinder engines, it will reach a practically constant flow condition through the filter.

Pressure will drop to some degree in all air cleaners. This pressure loss at the engine air inlet may not be objectionable under part-throttle conditions. But at full throttle, a measurable power loss may result if the air cleaner has a high pressure drop.

Care should be taken to learn the initial or clean filter pressure drop, average pressure drop over the period between servicing, and the pressure drop just prior to servicing.

Installation of air cleaners requires consideration of more than merely space requirements and physical clearances. Accessibility for servicing and air inlet conditions also must be checked. Intelligent consideration of air supply to the cleaner inlet may improve cleaner efficiency and extend time to overhaul.

For example, on mobile installations use of louvers or hooded air inlets ahead of the cleaner, placed to cause the air to turn from its normal path before entering the cleaner, has improved filter operation. In this way much of the heavier dirt it thrown out of the airstream when the air makes the turn into the cleaner.

Proper air flow distribution over the entire filter face area of the cleaner is essential. Evenly distributed air flow can be affected by the design of the cleaner itself or by the installation. A test should be made to assure a good air flow pattern.

Filter selection also must depend on the type and quantity of dust or dirt. Although certain standard dusts are used by the filtering industry for testing, efforts are being made to establish standard test methods to enable engine manufacturers and operators to rate better different types of air cleaners.

Dirt-holding capacity of the cleaner will establish the servicing period required. The operator must determine this period by observing the cleaning job and the increase in pressures loss through the cleaner as dirt loading progresses. Some cleaners will show an increase in cleaning efficiency as the dirt load builds up. But this also brings a higher pressure drop. Thus the operator should examine both efficiency and pressure loss curves of any cleaner to compare different types properly.

Initial cleaner cost must balance against required maintenance to determine true economics of air cleaning. (Paper "Air Cleaners for Diesel Engines," was presented by James Myers at SAE West Coast Meeting, Portland, Oreg., Aug. 16, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Plane Engines Compared On Operating Cost Basis

Based on paper by

HUGH M. HENNEBERRY

and A. F. LIETZKE

National Advisory Committee for Aeronautics

COMPARING air transport powerplants with operating cost per payload ton-mile as the criterion yields results including most significant factors over a wide operating range.

To apply this criterion to engine evaluation, the total cost per ton-mile must be expressed as a function of the operating variable. An equation ex-

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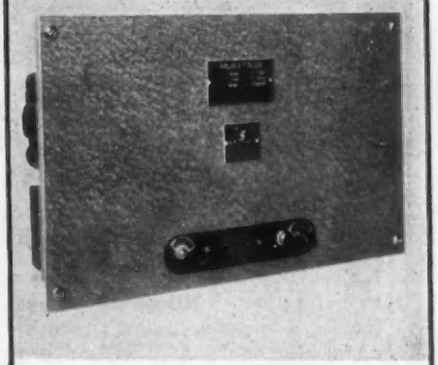

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Regulation Accuracy	0.3% from 0.1 to full load.				
Ripple Voltage RMS Maximum	1%.				
Recovery Time	0.2 seconds-value includes charging time of filter circuit for the most severe change in load or input conditions.				

*Adjustable + 10% —25%.
**Individual models identified by indicating output voltage first then amperes.
Example: 6-6-5 = 6 VDC @5 amperes.

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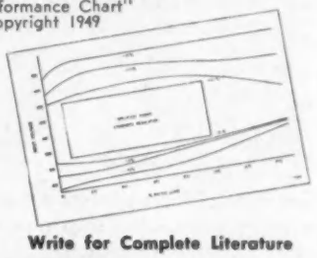

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pressing this function can be developed by considering the costs in three separate categories—direct, payload, and overhead.

Equations are available for expressing the first two items. But complete engine comparison hinges on the degree of accuracy attained in expressing the overhead costs. Even if the overhead costs cannot be rigidly evaluated, important conclusions can be drawn from an engine evaluation based on the total cost of operation per payload ton-mile.

Because the performance of an engine-airplane combination is greatly affected by the flight altitude, any powerplant evaluation should analyze each engine-airplane combination at the altitude giving best performance. The total operating cost per ton-mile therefore need only be determined at the optimum altitude.

Optimum altitude can be rapidly approximated by using proper simplifying assumptions without great sacrifice in the accuracy of the final results. (Paper "A Method of Evaluating Powerplants for Use in Subsonic Transport Airplanes," was presented at SAE National Aeronautic and Air Transport Meeting," New York, April 13, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Spacious Interior Graces Stratocruiser

Excerpts from paper by

D. B. MARTIN

Boeing Airplane Co.

ONE of the most impressive features of the Stratocruiser is its spacious interior.

The upper deck, two feet shorter than a standard Pullman, provides large dressing rooms, passenger compartments, and a galley. Wide, deep-cushioned reclining chairs increase comfort. For added relaxation, a 14-seat lounge and snack bar on the lower deck, connected by a modernistic spiral staircase, makes possible a change of surroundings during long flights.

The large cargo space—845 cu ft of it—offers operators additional revenue and versatility in operation. Large cargo doors at truck-bed height, for direct loading from ground carriers, obviate the need for loading ramps. A separate galley service door opens into the 350-cu-ft galley for direct loading. Cargo, mail, baggage, and galley provisions are loaded on the opposite side of the airplane from the main passenger entrance, thereby eliminating interference with passenger loading.

Interior arrangements vary with customers' fleets. Luxury day seating offers from 47-seat to 61-seat arrangements and numerous berthable-sit up arrangements. One airline has provided a "honeymoon compartment" in the aft part of the upper cabin. Another uses 39 Sleeperette-type seats and 17 berths for North Atlantic service. Several customers have a preference for a centrally located galley.

The Stratocoach version of the Stratocruiser seats 103 to 114 passengers. Rearrangement of the dressing room space and conversion of the galley area into seating area makes possible the high-density seating. The lounge remains unchanged. (Paper "The Present and the Future of the Boeing Stratocruiser" was presented at SAE Metropolitan Section on June 16, 1949.)

Engine Development Testing

Excerpts from paper by

EMMETT W. BOND

Ford Motor Co.

A NEW engine design goes through a basic testing program on the dynamometer to determine its performance and durability characteristics.

The engine first runs through a break-in period to insure frictional stability during development tests. The break-in consists of a predetermined schedule of initial low-speed, light-load steps, gradually progressing to high-speed, full-load steps.

After the running in, first step is to determine how closely the engine approaches the designed specifications originally set forth. The preliminary full-throttle test may be run with an approximate size carburetor, which is altered to provide a manual means for adjusting the fuel flow. This gives the very first picture of the new engine's performance.

From results of this preliminary power run emerges a program for determining exact carburetor size. Carburetors of various sizes are installed and manually adjusted so that a family of curves may be obtained, depicting effect of each on the engine's performance.

After finding the right size, it is necessary to engineer the carburetor

so that it will automatically meet the engine's fuel demands. Calibration here meets full throttle demands; additional adjustments for part-throttle operation may be necessary later.

Development testing that follows may vary in sequence. But to provide an overall performance picture of the new engine design, a minimum testing program would consist of the following investigations:

1. Manifolding,
2. Volumetric efficiency,
3. Heat rejection,
4. Torsional vibration of the crankshaft,
5. Motoring compression,
6. Spark advance requirements,
7. Endurance or durability, and
8. Miscellaneous tests.

Study of the manifold and its ability to distribute an equal air-fuel ratio to each cylinder is necessary. Probably no other one design factor is more dependent on development testing results than the induction system. Such an investigation usually includes two separate studies: (1) full-throttle operation and (2) part-throttle operation.

Fuel distribution is determined by comparing combustion chamber temperature as obtained for each individual cylinder. These temperatures are measured by a specially-developed spark plug in which a thermocouple is embedded. The thermocouple is located within the combustion chamber to get a representative temperature.

The carburetor on such a test is manually controlled to vary the air-fuel ratio. Several steps on the rich and lean sides of the best power fuel setting are run to get the cylinder temperature trend. Since best power and fuel mixture coincide with maximum temperatures, a plot of spark plug temperature versus fuel setting for each cylinder present will present the fuel distribution picture.

Measuring the amount of air the engine consumes while in operation checks its ability to induct air with respect to its swept displacement (volumetric efficiency) and also determines how effectively this air is used to deliver power. Such tests are made at full throttle with suitable equipment being provided to measure air flow into the engine.

Engine heat rejection is one of the tests performed only by the dynamometer. Purpose of this test is to determine the distribution of the heat provided to the engine in the form of fuel. This distribution normally divides into four parts: (1) heat to brake power, (2) heat to friction power, (3) heat to cooling water, and (4) heat to balance, such as exhaust products, radiation, and oil heat.

While the amount of heat converted to brake power and the amount lost to friction are readily determined by the dynamometer, additional equipment is needed to ascertain the heat to the cooling water. This may be done by locating a heat exchanger with the

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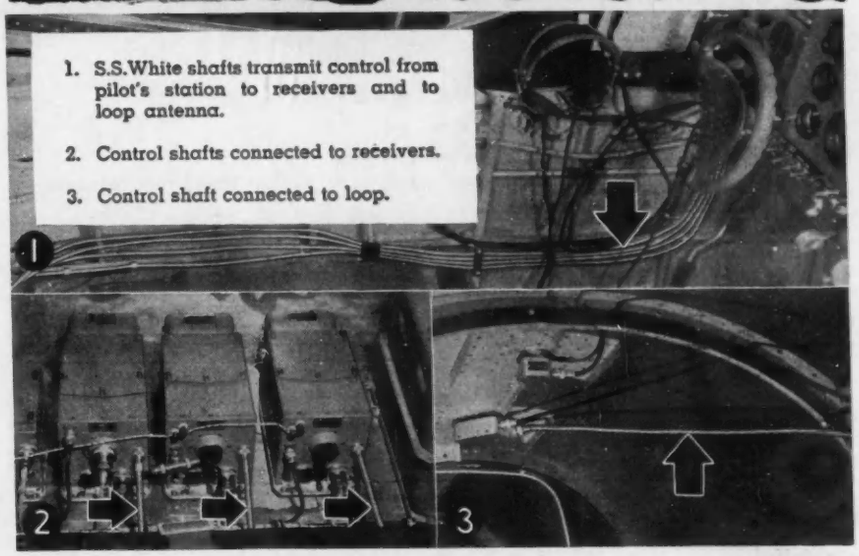
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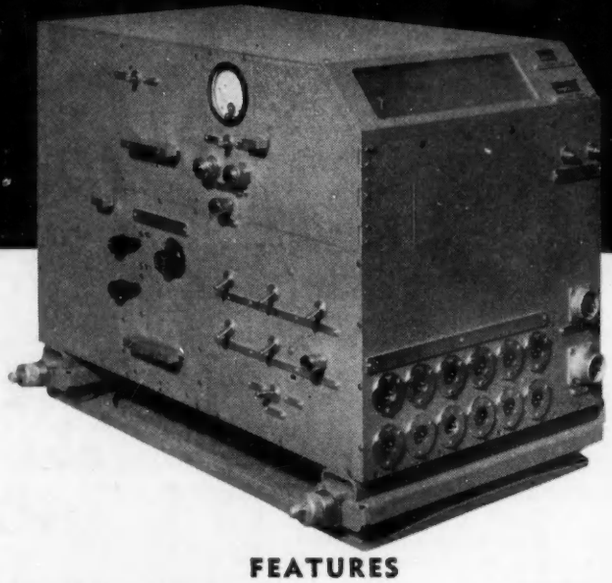

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necessary associated equipment within the engine cooling system. Besides the heat rejection data obtained, the engine water pump flow also can be calculated from the results of this test.

Another test which may most conveniently and accurately be performed on the dynamometer involves the determination of crankshaft torsional vibration. This rhythmical twisting of the crankshaft around its longitudinal axis is induced by the discontinuous flow of power impulses from the connecting rods to the crankshaft.

Whenever this impulse frequency or one of its harmonics about equals the torsional "natural frequency" of the crankshaft system, the crankshaft responds by vibrating torsionally with a measurable increase in amplitude. A torsional pickup located at the free end of the crank transfers the vibration signal through the electronic equipment to record the amplitude.

This test is performed on a new engine with a "bare" crankshaft—without a damper. If the amplitude is considered excessive, then a damper must be designed to reduce the peak amplitude values.

One of the most important test phases involves getting spark advance requirements, ignition timing sensitivity, and detonation characteristics of an engine over the usable range of speeds, loads, and mixture ratios. Tests at full and part-throttle conditions are performed to give the overall picture. Observed data from this test provide the ignition engineer with enough information to design an automatic distributor to fulfill requirements of the engine.

A motoring compression test, conducted early in the engine development program, indicates the relative air change among the cylinders and shows any ram effects. It approaches the induction problem from a different aspect than fuel distribution. The engine's ability to draw air into the cylinder is simpler than handling both fuel vapor and fuel.

When engine development approaches the final stages with overall performance determined, additional testing data on durability of engine components are essential. The investigation is carried on by running the engine at a high speed and full load. One typical endurance test calls for 100 hr full load at the predetermined limiting speed of the engine.

Additional dynamometer testing normally consists of investigating such items as oil consumption at various loads and speeds, piston blowby, air silencer development, muffler systems, and total accessory horsepower to predict "as installed" torque and horsepower. (Paper "Engine Testing," was presented at SAE Detroit Section, May 2, 1949. This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

New Members Qualified

These applicants qualified for admission to the Society between Aug. 10, 1949 and Sept. 10, 1949. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (Aff.) Affiliate; (SM) Service Member; (FM) Foreign Member.

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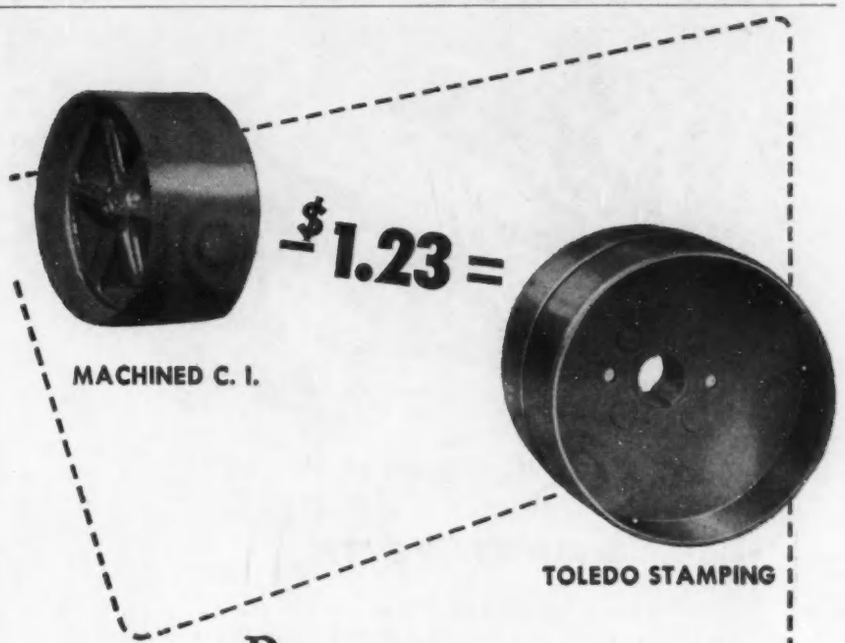

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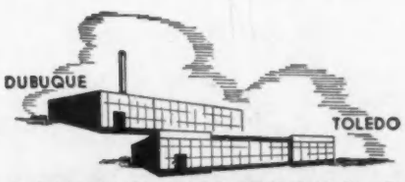

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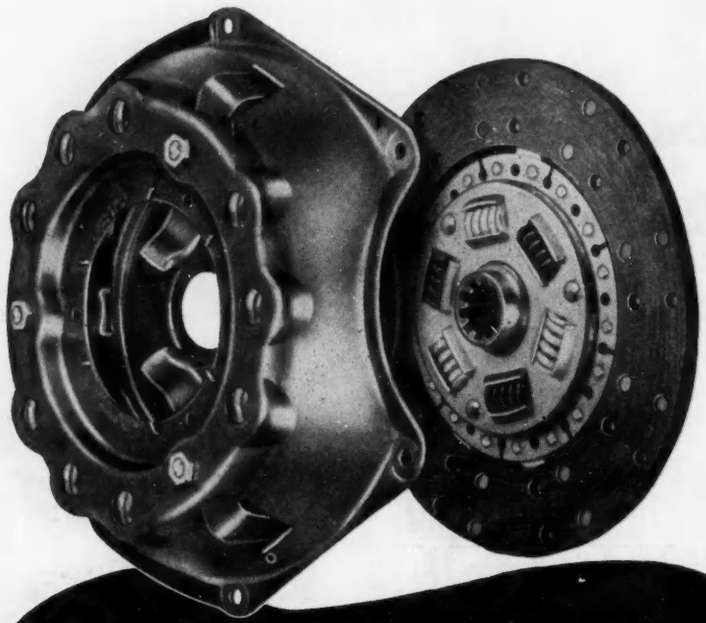

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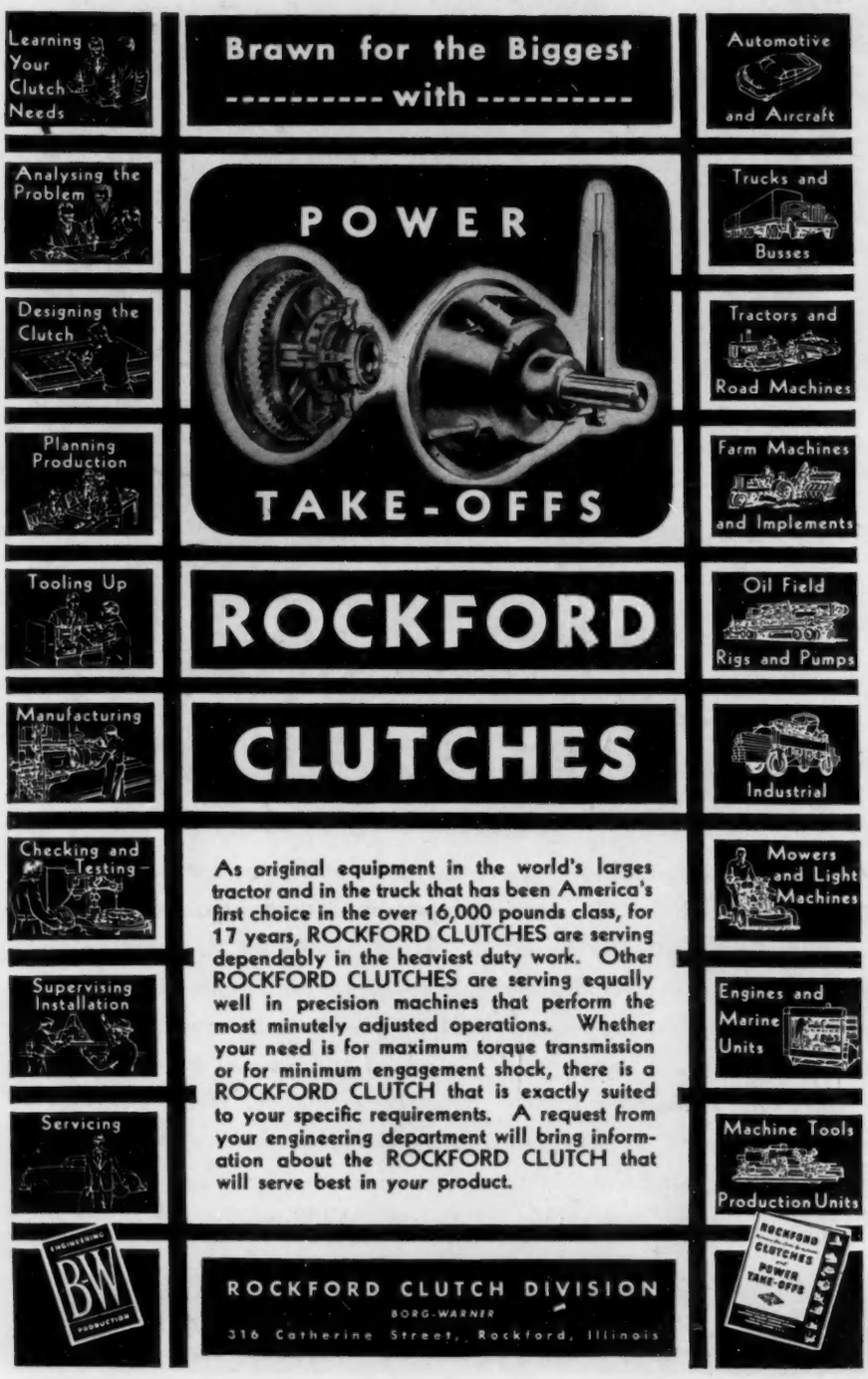
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